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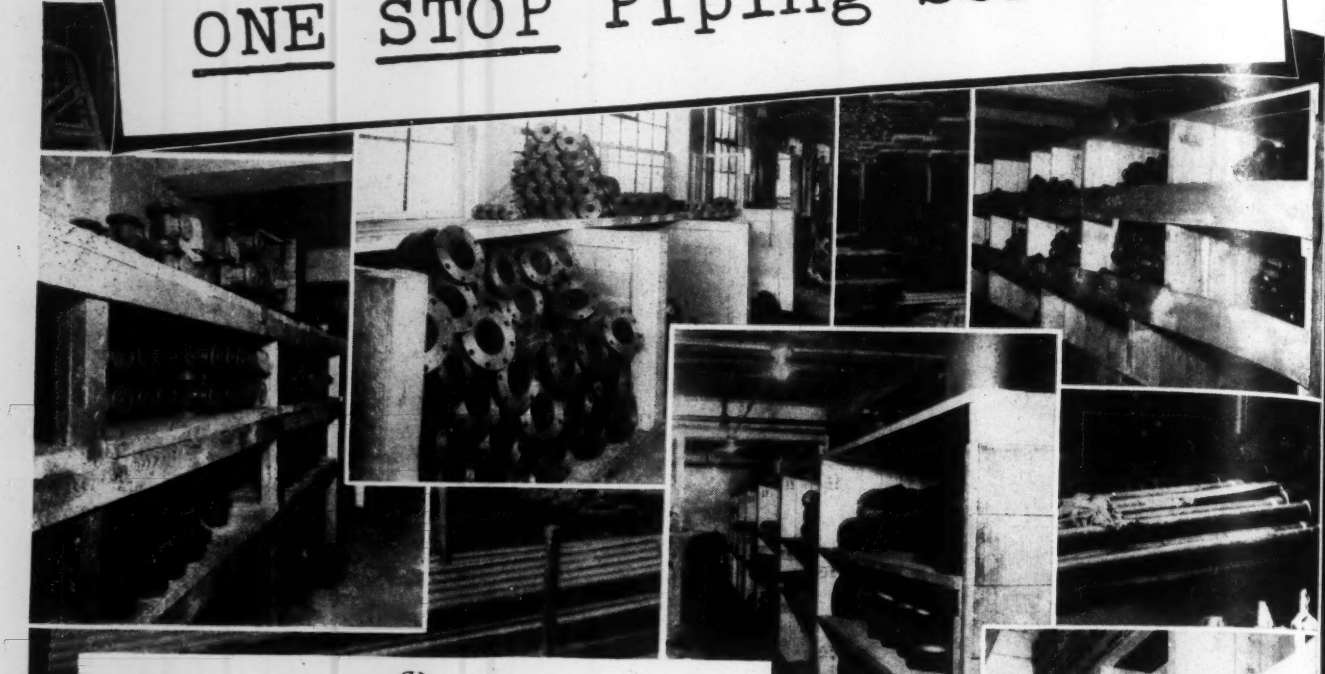

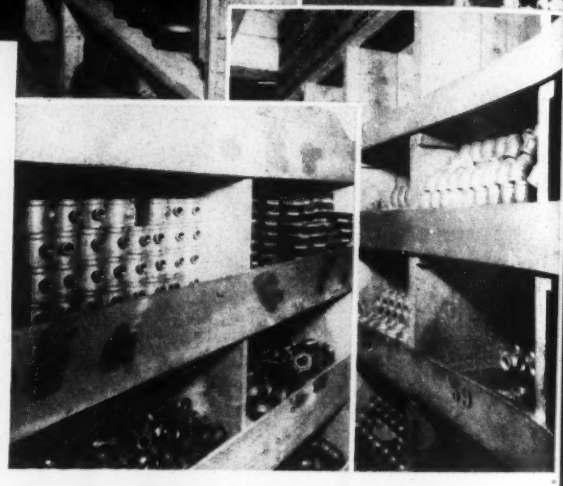

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THE EDITOR'S PAGES

ROLE OF PIPING IN INDUSTRY GROWS

Industrial expansion and technological change are both creating a tremendous increase in the use of fabricated steel piping and tubular metal goods, according to William P. Witherow, president of Blaw-Knox Co., in an address before the Pipe Fabrication Institute at its annual meeting in Pittsburgh.

There has grown an awareness of the shortage of tubular goods associated with the handling of liquid and gaseous fuels. But this is only one of the important reasons, Mr. Witherow reported, for the demand pressing on the piping industry. "Actually the growing demand for material of this type," Mr. Witherow said, "is tied to a long range trend and reflects a widespread change throughout industry in its use of integrated piping systems as a means of materials handling in continuous and mechanized processes.

"We must not assume, therefore, that the tight situation in this field is a 'flash in the pan,' or that it depends entirely on the backlog for the petroleum and gas industries. The role of piping is growing throughout all industry."

AIMS TO IMPROVE SPECIFICATION WRITING

A new organization dedicated "to the improvement of specification writing and practices in the construction and allied industries" has been launched; it is known as the Construction Specifications Institute, Inc., 1825 K St., N.W., Washington 6, D. C., and " is composed primarily of specification writers, architects, and engineers both in private and public practice."

Object of the institute is to compile and analyze statistics and information; to engage in the research and study of new methods and all problems dealing with specification writing; and to establish and maintain a clearing-house of unbiased technical information on specifications for the fabrication and installation of construction materials and equipment.

Priority goals set by the officers include these:

a) Better specification writing.

b) Simpler specifications.

c) Standardization of building codes.

d) Standardized specifications for public works by the government at all levels.

e) Study of new materials and other processes developed during World War II, in order to fit them into the construction picture as substitutes or improvements on costlier and scarce materials, "today retarding all government, residence, and commercial construction."

James B. Moore, chief of specification review unit, division of hospital facilities, U. S. Public Health Service, is president of the institute.

SPREADER STOKER FIRING TO BE STUDIED

Representatives of eight spreader stoker manufacturers and Bituminous Coal Research, Inc., met last month to study means for improving the utilization of bituminous coal. This meeting was brought about "by the increasing acceptance of spreader stoker firing in industrial and public utility plants and a desire to extend its versatility and overall efficiency."

A spreader stoker research committee was formed to administer the program, with William S. Major, BCR development engineer, as chairman. Plans were formulated to conduct research tests on a representative size and type of spreader stoker fired boiler under the supervision of the fuel engineering staff of Battelle Memorial Institute, Columbus, Ohio.

$3 BILLION FOR POSTWAR FACTORIES

American industry has invested approximately $3 billion in manufacturing plant expansion and new industrial buildings in the 37 states east of the Rocky Mountains since the close of World War II, according to an analysis made by F. W. Dodge Corp.

The actual dollar volume of contracts awarded for manufacturing building for the 2½ years covering the last quarter of 1945, the full years 1946 and 1947, and the first quarter of 1948 was $2,725,856,000, the corporation reported on the basis of project awards. The figures are for structures only and exclude processing machinery and equipment contained in buildings.

Processing industries, such as chemicals, rubber and textiles, accounted for $1,595,363,000 in building contract awards during the 30 month period, while the manufacturing industries, such as automobile and machinery manufacturing, accounted for the remainder.

BUILDING MATERIALS PRODUCTION INCREASES

Production of building materials to meet postwar construction needs has accelerated much faster during the last two years than it did following World War I, Melvin H. Baker, chairman of the Construction Industry Information Committee, said last month.

"A study by the committee's economists has revealed that the rate of increase in building materials output was three times as great in the first year following World War II as it was between 1918 and 1919," Mr. Baker said. "In 1947, the second postwar year, the rate of increase in production was almost twice as great as from 1919 to 1920, the corresponding period after World War I. The upward trend is continuing in 1948.

"It took only two years after 1945 to raise production by 50 percent, while it took five years after 1918 to increase output by a similar percentage.

"The rapid recovery of materials production from war conditions was in line with the fast expansion of general construction activity following World War II. The physical volume of all new construction activity, the committee pointed out, has more than doubled its wartime low, and in 1947 exceeded by 10 percent that of 1939, the last prewar year unaffected by war demand.

"This record shows the great flexibility of the construction industry in responding to demand. For a good part of the postwar period, nonresidential construction was subject to rigid governmental limitations. This fact makes the recovery record even more impressive."

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THE EDITOR'S PAGES

PIPING FIRM REPORTS GREATEST SALES VOLUME

Midwest Piping & Supply Co. had net sales of $19,169,666.49 for the fiscal year ended February 29, 1948, it was disclosed in the annual report issued last month. The company, which fabricates piping and produces welding fittings for utilities, refineries and other major industries, reported sales volume greater than for any other peacetime year.

Net profit was $1,895,966.13, the equivalent of $4.91 per share of common stock and 9.89 percent of volume of sales.

The firm's construction division installed piping for utilities and other major industries in 27 states in all sections of the country, the report states.

EXCISE TAX ON AIR CONDITIONERS

According to the Air Conditioning & Refrigerating Machinery Association, a ruling from the acting commissioner of internal revenue on section 3405(c) of the internal revenue code removes liability for the 10 percent excise tax from store coolers, leaving as taxable only equipment of less than 1 hp size, or room air conditioners.

The ruling results from long negotiations with the Bureau of Internal Revenue by ACRMA's tax committee, according to William B. Henderson, executive vice president of the association.

BUILDING CONTRACTS SET QUARTER MARK

F. W. Dodge Corp. reported recently that a new first-quarter high mark in dollar volume of construction contracts was set in the 37 states east of the Rocky Mountains with a total of $1,986,936,000. This volume surpassed by 23 percent the previous first-quarter record established last year.

While the dollar volume of residential contracts was maintained at a level equal to the first quarter of last year, nonresidential contracts were up 42 percent and heavy engineering awards increased 46 percent over the comparable period of 1947.

Thomas S. Holden, president of the corporation, in an interpretation of the activity record pointed out that the sharp increase in construction costs during the last year accounted for some, if not all, of the dollar volume gain.

The physical volume of nonresidential space as measured by square foot floor area and covered under contracts awarded in the first quarter was substantially in excess of that reported for the corresponding period of last year, Mr. Holden said. At the same time, considerably less new floor area is reported for residential buildings.

STORED NATURAL GAS IS DEHUMIDIFIED

How one well known gas utility is utilizing dehumidification equipment in an interesting natural gas storage operation was announced last month. Like other gas utilities faced with increased demands for its product, the Public Service Co. of Northern Illinois has long recognized the desirability of some type of natural gas storage with a sufficient capacity to help supply peak loads for space heating on cold days. After studying various storage methods, the company decided on storage at high pressure in buried pipe [see HPAC, May 1948, pages 93-97].

To insure the success of high pressure, underground storage from the standpoint of preventing the formation of hydrates the company installed activated alumina dehumidifier equipment to dry the field supply gas. The gas reaches supersaturation when compressed to high pressures in the storage field. Moreover, upon releasing the gas from the storage field and reducing its pressure to that of the distribution line a considerable refrigeration effect results. Hence, if the compressed gas is wet at high pressures and normal temperatures the refrigeration effect on expanding might freeze lines, valves, and other equipment.

According to calculations the maximum dew point that can be tolerated in this particular type of storage and at high pressures is —40 F before compression. It is also believed that dew points lower than —40 down to as low as —60 deg offer further advantages.

At Public Service's station near Mt. Prospect, Ill., operating data are as follows:

Odorized field supply gas is received at approximately 60 psig at present but within the next year this pressure is expected to reach 150 psig. The gas enters the dehumidifier at the supply line pressure and is dried to dew points of —40 F or lower. Passing through the dehumidifier, it is piped through a reducer regulating valve where it is regulated to the desired suction pressure of the compressor. Present suction pressure is zero but will later vary as high as 60 psig maximum. The gas then enters the compressor and is discharged into the manifold leading to the underground storage units which are manifolded together and have an ultimate storage capacity for each group of 1 million cu ft. All or part of the manifolded groups may be filled at once, depending on the number which are open to the compressor manifold by means of their own individual valves. Gas is supplied to the storage pipes until the final pressure of the group being filled is 2240 psig.

When the stored gas is ready for use during periods of peak demand it is run back to the station and through a steam heated heat exchanger which raises its temperature to 150 F at 2240 psig. This is done to relieve some of the tremendous refrigeration effect when the gas is later expanded. The heated gas is then reduced to the pressure of the distribution lines which at present is 60 psig, but will later be in the range of 125 to 150 psig. At this reduced pressure the gas is metered, re-odorized and distributed. Re-odorizing the gas is necessary because in dehydrating the gas the adsorbent contained in the drying equipment removes a substantial amount of the odorizer.

The dehumidifier built for the Mt. Prospect storage field is an electric reactivated machine and is equipped with a closed circuit reactivating system.

Heating Piping & Air Conditioning

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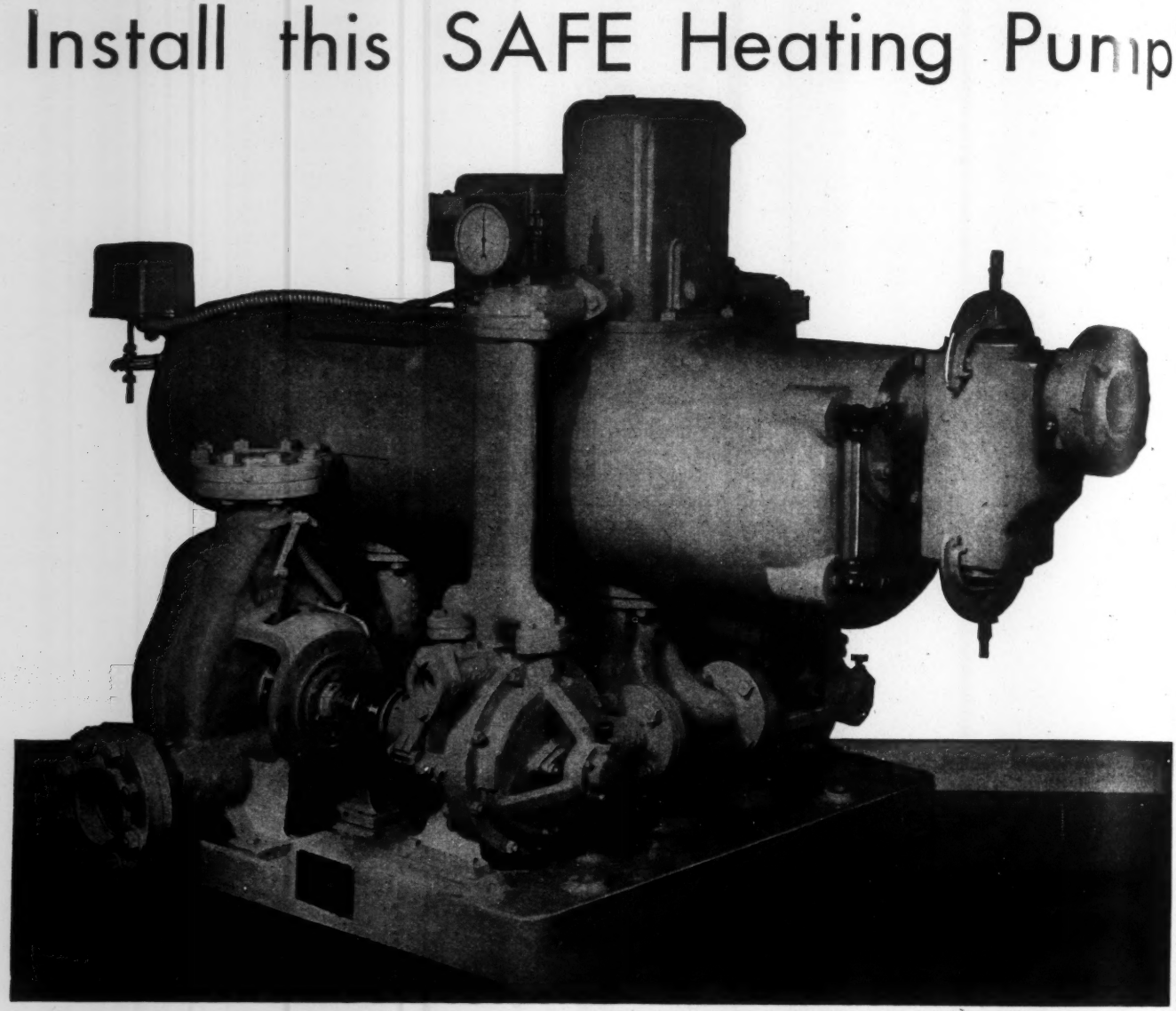

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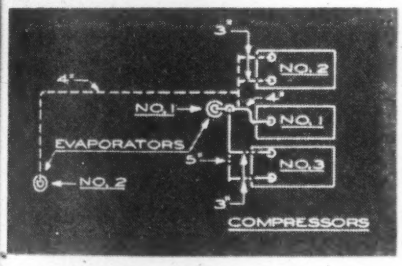
Improved Piping Design Aids Refrigerating Plant Results

HERMAN VETTER, consulting engineer, Los Angeles, shows here—in an easy-to-read article describing an actual example—how improperly designed piping in a refrigerating plant results in low efficiency, and how performance and capacity can be improved. Adding compressors or evaporators and increasing speed of the compressors without first checking the capacity of the existing piping are among the basic causes of inefficiency

A GREAT DEAL of attention is given to selecting efficient machinery in plant and building design. Often, however, too little attention is given to the connecting links (piping), with the result that the overall plant efficiency suffers. This is true whether it be an electric generating plant, steam power plant, or refrigerating plant. My purpose here is to show how improperly designed pipe lines in a refrigerating plant resulted in low plant efficiency.

The suction lines as they were before they were changed are shown in Figs. 1a and 1b. In the diagram, Fig. 1b, the various stages of the pipe installations are indicated by different types of lines; the lines for the first installation which connected evaporator No. 1 with compressor No. 1 are shown in full lines; the second stage, when evaporator No. 2 and compressor No. 2 were added, is shown in broken lines; the third stage, when compressor No. 3 was added, is shown by dash and two-dot lines. Both of the evaporators were of identical size and identical design.



The initial analysis of this plant indicated that the compressor capacity and the evaporator capacity were not in balance. The first step, however, in the rebuilding of this plant was to correct the piping and this article is confined to this phase only.

The first step in this analysis was to check the displacement of the compressors:

Compressor	Rpm	Piston Swept Volume per Rev.	Volume per Min
No. 1	104	1.965 cu ft	204.36 cfm
No. 2	124	2.06 cu ft	255.44 cfm
No. 3	180	2.48 cu ft	446.40 cfm

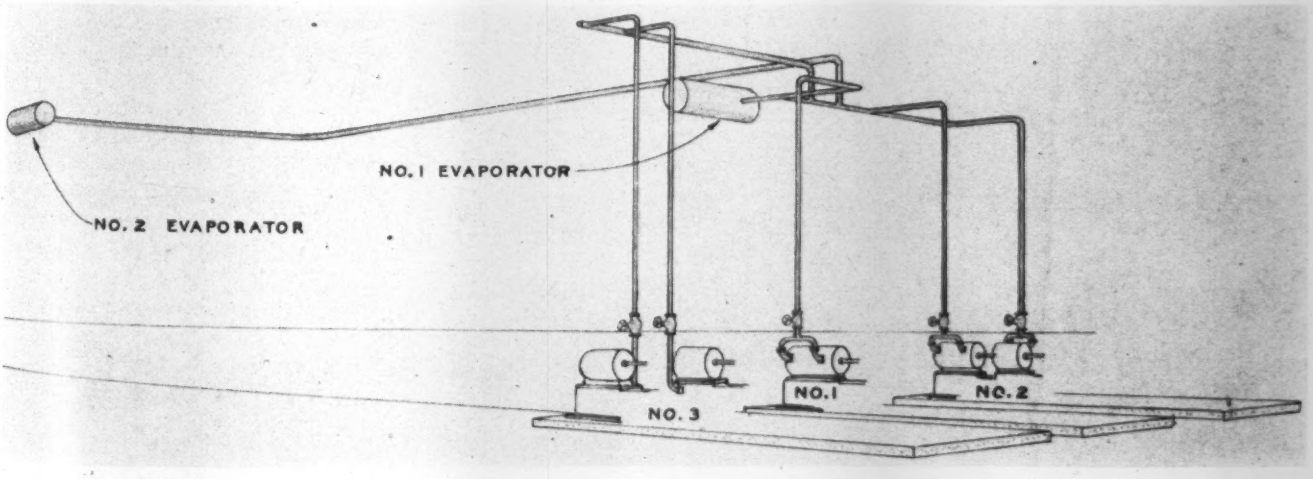
Referring to the piping diagram, Fig. 1b, it will be noted that compressor No. 3—which has almost as much displacement as the combined displacements of compressors Nos. 1 and 2—is at the end of the line. This arrangement allowed only that portion of the ammonia vapor not drawn into compressors Nos. 1 and 2 to flow to compressor No. 3.

The suction lines as they were after these lines were changed are shown in Figs. 2a and 2b, which also indicate where evaporator No. 1 was relocated after both of the evaporators were redesigned and the capacity enlarged. The test given in this article was made before the evaporators were enlarged.

Excessive Vapor Velocity Cause of Inefficiency

If the compressors were operating with a volumetric efficiency of 75 percent (which would be normal for this type of machine) the vapor velocity in the 4 in. lines from the evaporators would be 3844 fpm.

Fig. 1a (bottom)—Arrangement of the suction piping in the inefficient refrigerating plant. Fig. 1b shows this piping in diagram form. Fig. 1b (left)—Diagram of the piping before it was changed, the first installation being shown in full lines, the second in heavy broken lines, and the third (when compressor No. 3 was added) in the dash and two-dot lines



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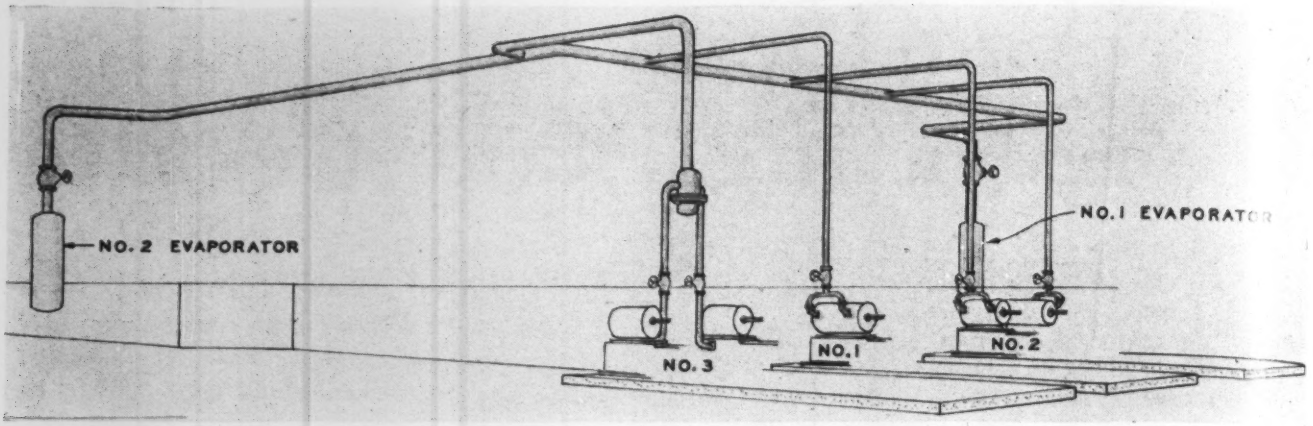
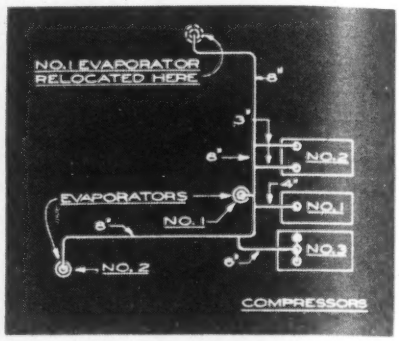


Fig. 2a (top)—Here is the suction piping after it was changed to improve plant performance. This piping is shown in diagram form in Fig. 2b. Fig. 2b—In the improved piping layout, an 8 in. suction line connects both evaporators, forming a header. With this arrangement, all compressors have equal access to the ammonia vapor



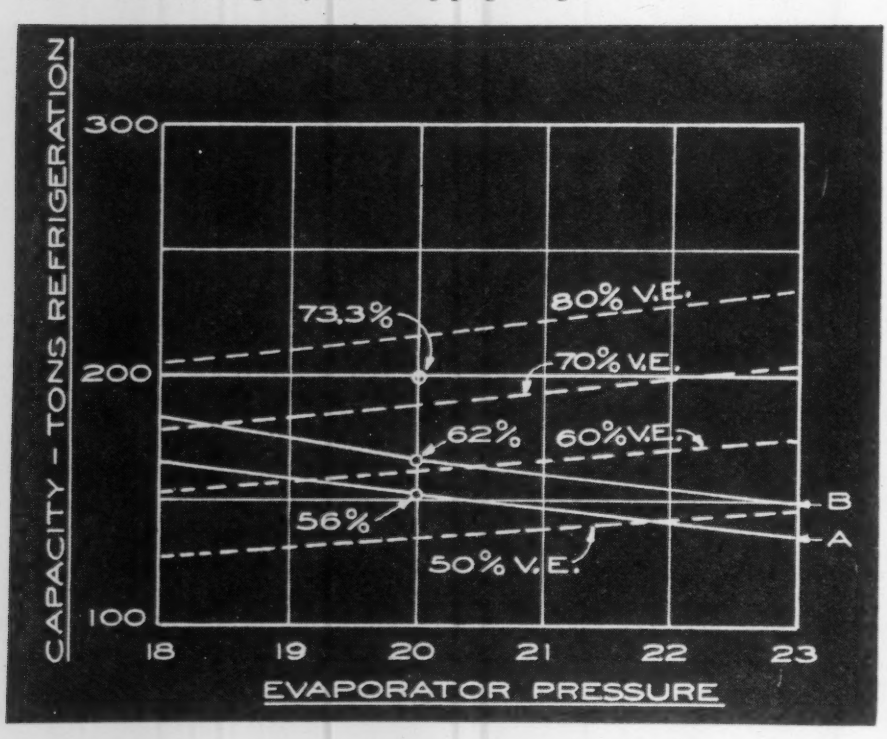
There are many who maintain that this is not excessive velocity; however, subsequent facts proved that it definitely was the cause of inefficient operation.

By referring to the piping diagram of Fig. 2b, it will be noted that an 8 in. suction line connects both evaporators, forming a header; with this arrangement, all the compressors have equal access to the ammonia vapor. The vapor velocity before the change was 2870 fpm and after the change the velocity was 805 fpm. These velocities are average and do not indicate actual operating conditions.

With reciprocating compressors, there is a large variation in the time-rate at which vapor is drawn into the compressor during each intake stroke. This is illustrated in Fig. 4. During the first quarter of the intake stroke, 17.4 percent of the vapor is drawn in; in the second quarter, 39.0 percent; in the third quarter, 31.6 percent, and in the fourth and last quarter, 12.0 percent of the vapor is drawn in. At the beginning and end of each intake stroke, the vapor velocity is zero and reaches its maximum before reaching the halfway point.

If the average velocity is 3844 fpm, the maximum velocity would be 6200 fpm. This of course would be the vapor velocity if there was only one compressor or if all the compressors were drawing in vapor at the same moment. Unless the compressors were interlocked and set so that the intake periods overlapped, the conditions mentioned above could occur quite frequently. At a recently built plant I visited, the pressure fluctuations in the suction line were so distinct that it was possible to count the compressor speed by timing the oscillations of the ammonia liquid in the gage glass at the evaporator.

Fig. 3—Compressor capacities for various pressures. Line "A" indicates combined capacity of both evaporators before the suction lines were changed, and line "B" shows the capacity after the piping change



In Fig. 3, the compressor capacities for pressures ranging from 18 to 23 psi and for 50, 60, 70, and 80 percent volumetric efficiencies are shown by broken lines. The full line "A" indicates the combined capacity of both evaporators before the suction lines were changed and were as indicated on the piping diagram, Fig. 1b. The full line "B" indicates the combined capacity of the evaporators when operated with the changed suction line as indicated on Fig. 2b.

After the suction lines were changed and before any changes were made in the evaporators, the capacity increased from 152 tons to 167 tons of refrigeration. This increase raised the average volumetric

efficiency of the compressors from 56 to 62 percent.

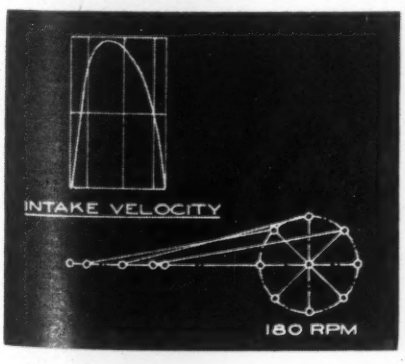
After the evaporators were enlarged and the No. 1 evaporator relocated as indicated in Figs. 2a and 2b, the capacity of the evaporators increased to 198 tons of refrigeration which also raised the average volumetric efficiency of the compressors to 73.3 percent. These volumetric efficiencies were determined from actual evaporator capacities and not from indicator cards. The evaporators were increased to the limit of the space available.

As the volumetric efficiency indicates the percentage of capacity derived from the compressor, the piping should be designed to give the highest efficiency possible. It should be realized that a compressor with a theoretical capacity of 100 tons will produce only 75 tons when operated with a volumetric efficiency of 75 percent.

The two basic causes of low volumetric efficiency are pressure drop due to line velocity and superheat. These are inter-related, because pressure drop represents power loss due to friction and friction causes superheat. In order to illustrate the effect of superheat, Fig. 5 is shown.

This diagram was prepared with the assumption that the compressor operates at 75 percent volumetric efficiency when the suction vapor enters the cylinder at saturation or zero degrees superheat. The ideal operating condition would be for the vapor to enter the cylinder at saturation. This, however, is seldom practical because there is too much danger of carrying slugs of liquid refrigerant over to the compressor. For good practical operation, the plant should be designed to operate with 10 deg superheat.

Fig. 4—With reciprocating compressors, there is a large variation in the time-rate at which vapor is drawn into the compressor during intake stroke



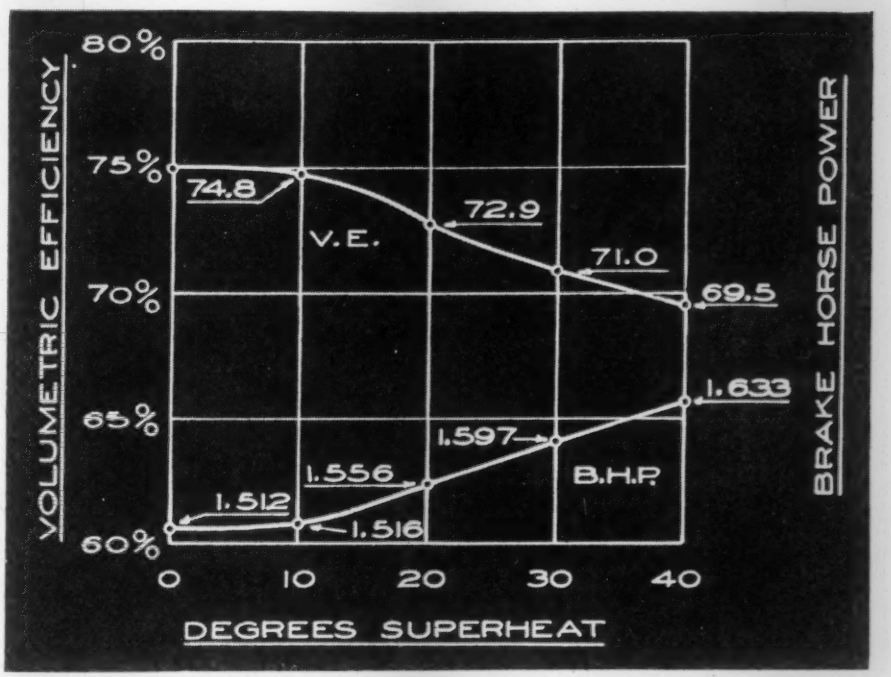


Fig. 5—The two basic causes of low volumetric efficiency are pressure drop due to line velocity and superheat, and these are inter-related

By referring to Fig. 5, it will be noted that not only does the capacity of the compressor decrease with a rise in superheat but the power per ton of refrigeration increases. A railroad company that owns and operates large ice making plants limits line velocity to 1200 fpm. This is good sound practice because the public utility furnishing the power is the only one that benefits by high vapor velocities.

At the plant which was used as the basis of this article, the superheat of the vapor entering compressor No. 3 was so high that liquid ammonia had to be expanded into the cylinders to keep them from overheating. This was entirely eliminated after the suction lines were increased in size.

Many Plants Could Benefit by Piping Changes

In conclusion, let us consider how these inefficiencies develop in plants similar to that analyzed in this article. By no means is this a rare case, and based on my more than 30 years of experience I can safely say that more than 50 percent of existing plants could have their efficiency increased simply by changes in the piping systems. The most extreme case that came to my attention had its power consumption reduced 75 percent.

Referring again to Fig. 1b, it will be noted that the original installation was fair in design for efficiency; the second addition neither increased nor decreased the efficiency of the plant; the third addition, however, caused a big drop in efficiency. Making additions to an existing plant without making a thorough engineering analysis of the entire plant is the underlying cause of most inefficiencies in plants.

It is not possible to make an engineering analysis of such plants with the fee based on the construction cost of the change. For instance, in the plant under discussion, the total cost of the change was $1800 without crediting the salvage of materials removed. The increased net income effected by these changes was $3400 per year. The engineering fee should therefore be based on the savings effected and not on the cost of the changes. Unless this is done, insufficient time is given to the analysis and the result achieved is unsatisfactory.

The basic causes of inefficiency can be summarized as follows:

1) Adding compressors or evaporators or both without checking capacity of existing pipe lines.

2) Increasing speed of compressors without first checking capacity of pipe lines.

3) Careless engineering, or—as in many cases—dependence upon "free engineering."

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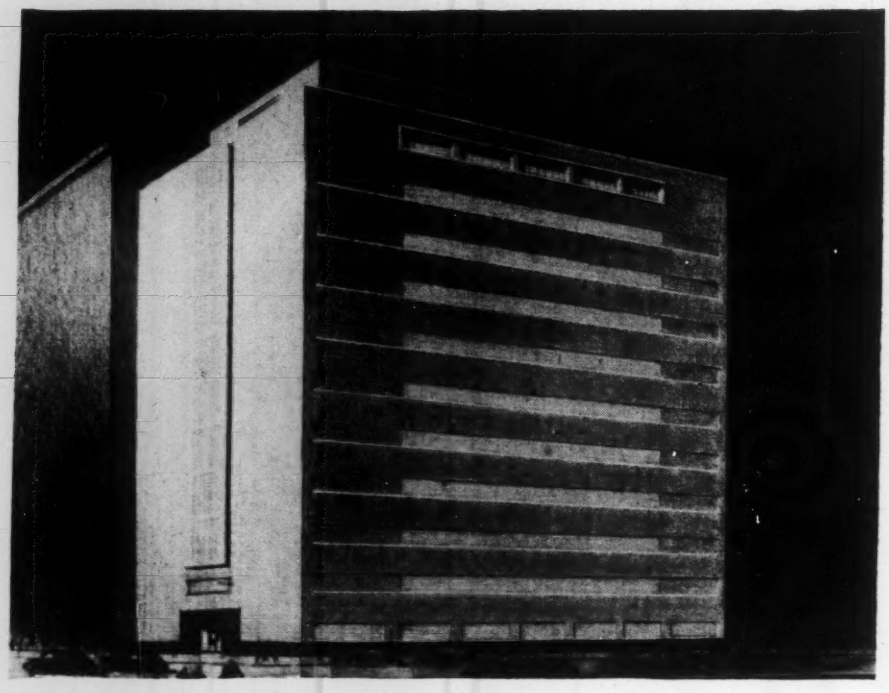

The new completely air conditioned No. 2 Toll building of the Illinois Bell Telephone Co., recently erected just south of the Chicago "Loop"

Big New Telephone Building Completely Air Conditioned

THE No. 2 TOLL building recently constructed at 85 W. Congress St., Chicago, for the Illinois Bell Telephone Co., is completely air conditioned.

All 11 floors are designed for telephone equipment. The floor to floor heights are 14 ft 6 in. and the floor slabs are constructed to carry a live load of 150 lb per sq ft. The exterior of the building is modern in design, faced with limestone and glass block panels for the street elevations and light face brick with a minimum of windows for the other two elevations.

At this time the first four floors have equipment frames and the fifth floor contains telephone switchboards, operators' quarters, and associated clerical space. The telephone power plant occupies about half of the basement. The first and second floors house the terminal equipment for the Chicago-South Bend-Cleveland and Chicago-Terre Haute-St. Louis coaxial cable systems and later on will contain the terminal equipment for the Chicago-Milwaukee radio relay system. These systems form a midwest network which now carries telephone messages and it is expected that it will be equipped to carry television programs some time this fall. It will probably be connected to the eastern network early in 1949. The third and fourth floors contain automatic toll switching equipment.

The sixth to 10th floors are finished for office space and a modern cafeteria and lounge are on the 11th floor.

The fifth to the 11th floors, inclusive, the elevator lobbies, and rooms off the elevator lobbies (rest rooms, washrooms, etc.) have suspended

HUMIDITY CONTROL is important for equipment performance because changes in relative humidity produce dimensional variations in certain materials employed in telephone apparatus. And filtering the air supply—particularly the outside air—is a "must" from the viewpoint of equipment performance, for there are literally millions of contact surfaces in a dial equipment room, and dirt settling on them may cause open circuits, corrosion, or resistance noises. . . . Therefore, the Illinois Bell Telephone Co.'s new No. 2 Toll building in Chicago, an 11 story structure, is completely air conditioned. The method of humidity control and filtering (which includes pre-filtering, electrostatic cleaning, and after-filtering) are briefly described here

ceilings and recessed fluorescent lighting.

A fan room is provided in the basement, on each floor and in the penthouse. Each fan room has a central system consisting of supply and exhaust fans, air filtration equipment, heating coils, and a dehumidifier casing containing a spray chamber and chilled water cooling coils.

Pre-Filters and After-Filters

The air filtration equipment consists of a pre-filter of 2 in. glass wool, an electrostatic filter, and an after-filter of 14 oz wool felt. The wool felt after-filter is installed mainly as insurance against failure of the electrostatic filter and to catch blowoffs.

Filtering of the air supply, particularly the outside air, is a "must" from the viewpoint of equipment performance. There are literally millions of contact surfaces in a dial equipment room and dirt settling on them may cause open circuits, corrosion, or resistance noises. Dirt accumulations on apparatus may interfere with mechanical operation of moving parts and where the dirt is abrasive in nature mechanical wear will be accelerated in bearings and on contact surfaces. Air filtration minimizes the equipment cleaning problem with the attendant hazards to wiring and apparatus adjustments.

Air Cooling and Humidity Control

The refrigeration system is in the basement and consists of two 225 ton centrifugal compressors, water cooled condensers and water coolers. "Freon-11" is the refrigerant. Two sets of duplex chilled water pumps are provided. One set and associated piping serves the basement and the first to fourth floor systems. The other set serves the upper floor systems.

The central systems on the first four floors have cooling coils sized to maintain temperatures at 80 F with 45 percent relative humidity when the outside temperature is 95 F. Temperature in warm weather is maintained at 75 to 85 F, depending on variations in outside temperature from 75 to 98 F. When the relative humidity exceeds 45 percent, the air is chilled to remove the excess moisture and is reheated as necessary; also, the fresh air intake is reduced to a minimum even though the outside temperature may be in the range of 65 to 75 F. Manual dampers are installed above the dehumidifier casings to bypass a portion of the air.

During the heating season moisture will be introduced into the air stream by means of water sprays to maintain a minimum of 35 percent relative humidity.

Humidity control is important for equipment performance because changes in relative humidity produce dimensional variations in certain materials employed in telephone apparatus. Also, at the lower values of relative humidity the quantity of airborne dirt increases and this dirt tends to settle out on contacting surfaces.

The central systems serving the upper floors have cooling coils sized to maintain temperatures at 80 F with 50 percent relative humidity when the outside temperature is 95 F. Temperature in warm weather is maintained at 72 to 83 F depending on variations in outside temperature from 72 to 95 F. When the relative humidity exceeds 50 percent, the fresh air intake is reduced to a minimum even though the outside temperature may be in the range of 65 to 75 F. These systems also have manual bypass dampers above the dehumidifier casings.

Heating

A three drum, bent water tube, 225 hp low pressure boiler has been installed in the adjacent Wabash building boiler room in connection with extension of steam service to the No. 2 Toll building. The new boiler and one of the two existing 125 hp boilers were arranged for oil firing. A duplex vacuum pump was installed in the No. 2 Toll building basement.

A minimum of direct radiation is provided in the telephone equipment rooms. Sufficient direct radition is provided in the upper floors to care for the wall and window losses and are controlled by room thermostats set at 72 deg which are mounted on the outside walls. The larger part of the heat load is handled by copper finned nonfreeze type heating coils in the central systems.

The automatic temperature control system is of the pneumatic type with duplex high pressure compressors.

Exterior Walls Vaporproofed

The interior surfaces of all exterior brick walls have a vaporproof seal. In addition there is a 1 in. air space, 4 in. tile wall, and a ¾ in. plaster coat.

Windows are held to a minimum. Those provided are the conventional double hung type of steel construction and are weatherstripped. Interior wood sash are installed and are weatherstripped.

The roof construction includes a 2 in. thickness of cork set in waterproofing.

Special cantilever beam framing is provided in one bay the length of the building to give a clearance of 12 ft 6 in. under all ductwork with the floor to floor height of 14 ft 6 in.

Holabird and Root and Burgee were the architects and engineers. George A. Fuller Co. was the general contractor. Hanley & Co. was the heating and air conditioning contractor, and Jamar-Olmen Co. the ventilating contractor.

ARCHITECTS, CONTRACTORS TO COOPERATE

The American Institute of Architects and the Associated General Contractors of America, 1227 Munsey Bldg., Washington 4, D. C., have established a joint committee which will consider problems in building construction of mutual concern to architects, contractors, and the public. It will be known as the national joint cooperative committee of the AIA and the AGC.

For a number of years the two associations have worked constructively together on such matters as standard contract provisions, simplified specifications, and recommended bidding practices. Only recently the architects and the contractors jointly developed and published A Suggested Guide to Bidding Procedure which, in the interest of the public, established standards for obtaining bids that eliminate waste and misunderstanding that might lead to unexpected costs for the owner.

The national joint cooperative committee formalizes the successful working relationship between the two associations on mutual problems.

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This snow melting sidewalk at the shop and service building of the Peoples Natural Gas Co., Johnstown, Pa., uses sinuous coils of pipe connected to a small gas fired boiler. A switch controls the system, starting the circulating pump and causing the gas burner to ignite from a constantly burning pilot. The coils are spaced so that the output of the 80 ft by 6 ft sidewalk slab is about 100 Btu per hr per sq ft. This is not the installation described here

We Let Hot Water SHOVEL OUR SNOW

J. H. MOWREY, plant engineer at the Beech-Nut Packing Co., Canajoharie, N. Y., explains here the method of construction and the basis for design for the snow melting systems for the main entrances of the plant. These units have been in satisfactory and economical operation for two winters. Many plants, stores, and other buildings are expected to install similar snow melting provisions during the summer for use next winter

HOT WATER SHOVELS the snow from the walks of the main entrances of the Beech-Nut Packing Co. plant at Canajoharie, N. Y., and leaves them not only clean but dry as well. Heating grid installations beneath the concrete slabs, each of which is about 60 ft long, eliminate manual labor. These units have been in operation for two winters and the results are extremely satisfactory and very economical.

The units are placed in operation about November 1 each year and are operated continuously all winter since no antifreeze is employed in the circulating water. Continuous operation also eliminates any time lag due to walks becoming cold during intervals between snow storms.

Circulating water is heated to 150 F in a hot water generator by steam at 120 psi pressure. The temperature is governed by a thermostatically controlled valve. Each walk has its own circulating pump. The system is operated at atmospheric pressure, an expansion tank being used near the heat exchanger. Each grid has a vent cock located at the highest point.

The grid systems are so designed that any path followed by the water, from the time it leaves the supply line until it enters the return line (see Fig. 2), requires that it pass through equal lengths of a given size of pipe. For example, water passing through the top grid flows a very short distance through the feeder header, but through the full length of the header at the return end—whereas water flowing through the lower grid passes through the entire length of the feeder header and through a very short length on the return end. This tends to equalize pipe friction in all runs.

The entire pipe system was constructed of wrought iron pipe and is oxyacetylene welded throughout. All joints were tested hydrostatically to 125 psi. When the system was installed, it was so placed as to provide complete drainage in the event of pump failure during freezing weather. After installation, it was found that it would operate by natural flow.

The grid was installed on a subbase of crushed stone 5 in. thick and about 2 ft wider than the con-

crete slab (see Fig. 1). This subbase was flush with the top of the grid and flush with the top of the slab on either side. The construction provides a cushion for the grid, a certain amount of insulation between the grid and the soil below, and aids in drainage. The slab was poured 5 in. thick with a 6 in. mesh of No. 6 wire for reinforcing. Suitable expansion joints were included.

Operating Cost "Very Economical"

The cost of operation of the walks is very economical. Snow shoveling is eliminated. There is no necessity for the use of salt or sand as a nonskid. The walks are not subject to the usual deterioration due to frost, since they never freeze; neither is there any heaving in the spring.

The elimination of nonskid preparations means the walk is clean. This fact, in itself, means economy because these walks are at two of the main plant entrances and accommodate about 2000 employees each day besides numerous representatives of other companies. The dirty water and mud which previously was tracked into the plant created a sizable cleaning problem. Steam and electrical energy consumption is almost negligible. As stated above, steam is supplied at 120 psi pressure from a 2 in. main. During each cycle of operation, the steam control valve is open about 3 min, a part of which time it is only partially open.

This cycle repeats about three times per hour during the coldest months (December 15 to March 1) and as the average outside temperature rises, the interval becomes longer. The circulating pumps operate continuously and are driven by 1 hp motors.

How to Figure Heat Required

In the computation of the system, certain assumptions were made. It was assumed that:

10 in. of snow = 1 in. of ice.

1 cu ft of ice = 57 lb.

1.2 in. (0.1 ft) = maximum fall of snow per hour.

Ground loss = 40 percent.

15 F drop in temperature between supply and return.

Heat of fusion of ice = 144 Btu per lb.

As an example, assume we want to melt snow from a driveway 100 ft long and 10 ft wide.

From the above assumptions, 1.2 in. or 0.1 ft of snow must be melted per hr. This is the equivalent of 0.12 in. of ice per hr. Then 12 in. × 12 in. × 0.12 in. = 17.28 cu in. of ice must be melted from each sq ft every hr.

(17.28 cu in.)/(1728 cu in.) × 57 lb (weight of ice per cu ft) = 0.57 lb of ice to be melted per sq ft per hr.

0.57 lb × 144 Btu = 82, and 40 percent of 82 = 33 Btu loss to ground.

A total of 82 + 33 = 115 Btu is therefore the heat required to melt 1.2 in. of snow per hr from each sq ft of walk.

Then 115 Btu × 100 ft long × 10 ft wide = 115,000 Btu. This is the total amount of heat required per hr for a 100 ft x 10 ft driveway.

Pump and Pipe Sizes

Since a temperature drop of 15 F is allowed in the system, 115,000 Btu divided by 15 = 7667 lb of water, which is the quantity required per hr, or 128 lb per min, to accomplish the necessary work.

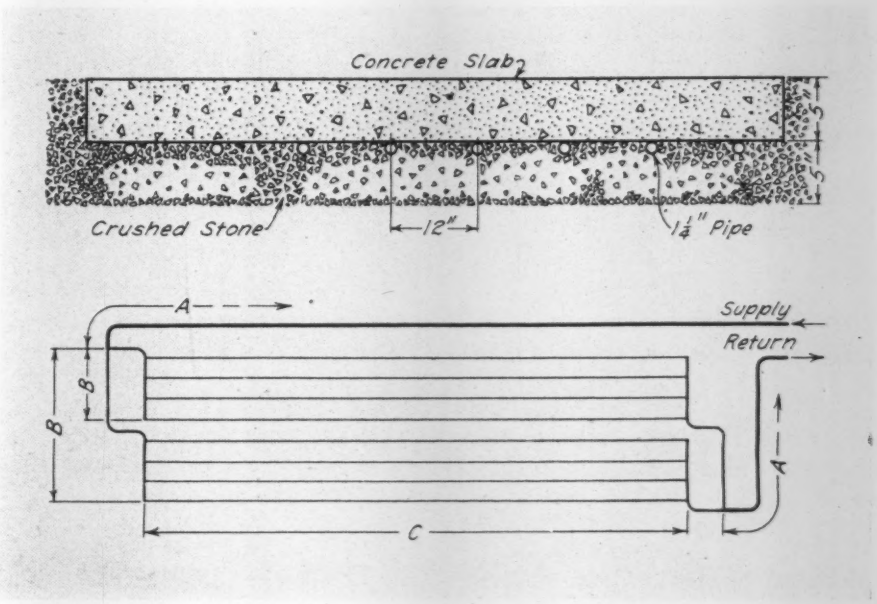
Since 1 lb of water = 0.12 gal, 128 × 0.12 = 15.4 gal of water per min is required to melt 1.2 in. of snow from a 1000 sq ft area. Then, 0.0154 gal will be required to melt this quantity of snow per sq ft per hr in this section.

From the above computations, it is evident that 1¼ in. pipe is not required for the quantity of water flowing. However, the radiating surface had to be considered as well as pipe capacity, and it was felt that 1¼ in. pipe would present an area large enough to care for this need.

The pumps used on the systems were taken from our stock. These pumps had larger capacities than needed and therefore the quantities of water used per minute were greater. However, on one walk where a 50 gpm pump was used, giving us three times the quantity of water required by computation, the temperature difference between supply and return averaged slightly under 5 F.

The operating costs of the two walks combined average about $1.60 per day. This figure includes fuel and electrical energy costs and boiler room labor averaged into coal costs. We feel that this cost is far less than the cost of keeping our lobbies and corridors free of tracked-in antiskid and ice melting substances used on most walks. So, because of these advantages, we let hot water shovel our snow.

Fig. 1 (top)—The pipe grids were installed on a subbase of crushed stone 5 in. thick and about 2 ft wider than the concrete slab. This construction provides a cushion for the grid, a certain amount of insulation between the grid and the soil, and aids in drainage. Fig. 2 (bottom)—The grid systems were designed so that any path followed by the water, from the time it leaves the supply line until it enters the return, is of equal length. The supply and return pipes marked "A" in this diagram of a heating grid are 3 in. pipe, the headers marked "B" at both ends are 2 in., and the grid itself ("C") is 1¼ in. pipe



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HOW TO SELECT FUEL

for Plants and Buildings

FRED MEYER, combustion engineer, Mid-West Heat Service, Chicago, discusses the perplexing (particularly these days) problem of fuel selection faced by the designing and operating engineer when planning a new boiler room. Flexibility in the fuel used must be provided for best results, circumstances often dictating that different fuels be used during different parts of the year ("dump" gas in the summertime, for example). A following article will discuss the subject of control rather thoroughly. Mr. Meyer presented a paper on this subject at a meeting on planning held by the American Hospital Association

FUEL SELECTION for a particular installation has always been a perplexing problem for the designing and operating engineer when planning a boiler room, and the postwar fuel situation has made it more complicated. It has been augmented by an "economy of scarcity" in fuels, and considerable thought in design and planning is indicated by today's conditions.

The first basic question is the relative cost of fuels. It is difficult, of course, to generalize on any statements of this type, because plants that may appear to be identical in design, construction, and operation face different fuel selection problems.

More fundamental than anything else is, first, the locality of the plant with respect to the availability of any particular fuel. It is obvious that a plant in the heart of a coal mining area would probably find that coal would be more economical than oil, gas, or a combination of these. Conversely, if the plant is in the heart of an oil refining area or in close proximity to a gas field, oil or gas may be more economical from the cost standpoint.

Selection logically follows along the lines of determining availability of fuel to the plant location selected, a study of the method of transportation of the fuel from its source to the facility, and the method of handling the fuel relative to unloading, storage, and use. Particular attention should be given to carload facilities, if this is a possibility, because in the case of coal or oil, the purchase of fuel on a carload basis is much more economical than on truckload basis in almost all cases.

Type of Load Affects Choice

The next important consideration is the size of the plant involved, the type of boilers in use or contemplated, and the type of steam load to be encountered.

In order to make the problem specific, let us pick an example of an installation in which we have a plant operating in the center of an otherwise residential area, with a peak load of approximately 400 hp and an operating pressure of 100 psi. Let us suppose further that the load varies during the severest conditions, from say 400 hp to 200 hp in 3 hr. Suppose further that we are in a coal producing area and that the load is to be carried by two 150 hp horizontal return tubulars.

Our first and only consideration would probably be coal, due to the proximity of the fuel and its low cost, and we might select either an underfeed or a spreader stoker for our firing equipment. For the type of load that we have imagined, an underfeed would probably be quite satisfactory, low in initial cost, and would produce an efficiency sufficiently high to make the selection of this type of equipment desirable.

Now, let us change our problem slightly. Suppose that we take the 400 hp to 200 hp drop in load in 30 min instead of 3 hr and then find ourselves back to the 400 hp peak at an interval approximately 30 min later, maintaining the load for, let us say, a 1 hr period. It is obvious that in this case a spreader stoker might now be a happier selection due to its ability to change load quickly and economically.

Unfortunately, our original premise included the fact that we were in an otherwise residential area and so, of necessity, the problem of fly ash must be considered. If fly ash is objectionable in this area, then the possibility that a spreader stoker might be undesirable must be considered.

If this same plant, with the same load conditions, is now placed in an area where coal and oil are equally available, with little or no cost differential (calculating cost of fuel against fuel only), we have created a new problem. Again, assuming a rapid variation from maximum to low load condition in a short period of time, we find that we would prefer to use a spreader stoker but hesitate because of the fly ash problem; in this event, we would examine carefully the possibility of using fuel oil. In this particular instance, cost of fuel has for the moment become a secondary consideration.

The problem of availability of the various fuels has caused many a

designing engineer to wonder what the proper recommendation should be. We have been facing the unusual condition of shortages in coal, oil, and gas throughout the post-war period. A close analysis of this situation indicates that all three shortages will eventually be overcome.

The shortage in coal is apparently due to two conditions: labor unrest, and lack of sufficient railroad equipment adequately to move large quantities of fuels required. The shortage of oil has been due primarily to four conditions: lack of wildcatting in bringing in new sources of oil; lack of refining capacity commensurate with new demands; lack of sufficient storage space to handle the increased demand for liquid fuel; and lack of sufficient transportation to bring this fuel to market.

Many engineers designing plants under these conditions have felt that in the ultimate, the design of any boiler plant should be so arranged that fuels may be interchanged, should this become advisable; and that any design not incorporating the possibility of such a change would not be considered good engineering. This does not seem to be an undue precaution, and the consideration of equipment arranged so as to make an interchange of fuels possible should be given thorough consideration in designing any plant.

Another important factor that must be considered in selecting a fuel is that of the storage facilities. With fuel supply as erratic as it has been in the past few years, most designers are trending toward large storage facilities. With this thought in mind, careful planning should be the keynote in designing storage for coal. If adequate storage room is not available, this may be a consideration for deciding that oil should be used, as it may be stored underground. Coal may be stored in the open but serious deterioration occurs when this is done, with consequent loss in Btu value of the coal.

We have already mentioned some of the problems occurring when fly ash exists. Cognizance should be taken of the fact that fly ash offers problems of cleaning buildings and equipment which may be in the vicinity of a stack exuding fly ash. In institutional buildings particularly, this is a serious problem. Some industrials also find difficulties with fly ash, due to the fact that the type of products they produce must be kept meticulously free from this type of contamination.

Fuel Handling in the Plant

In analyzing the operating costs of a boiler room after we have considered initial fuel cost, the next problem is the fuel handling in the plant itself. We must examine, in the case of coal, cost per ton to handle fuel from the storage position to the point of consumption. We must determine the cost of handling the fuel residue from the boiler to its point of disposal. We must obtain the operating cost of the firing equipment, maintenance costs for the equipment itself, refractory brickwork costs, and the cost of maintaining the boiler itself and

Fig. 1—Combination gas-oil burner, showing the burner from the outside of the boiler as it would be applied to a typical installation. The gas burner may be removed readily by breaking the union and opening four bolts, which removes the plate from which the gas ring is suspended. This entire assembly is removed and burner then swings into normal position. The same controls may used for either oil or gas

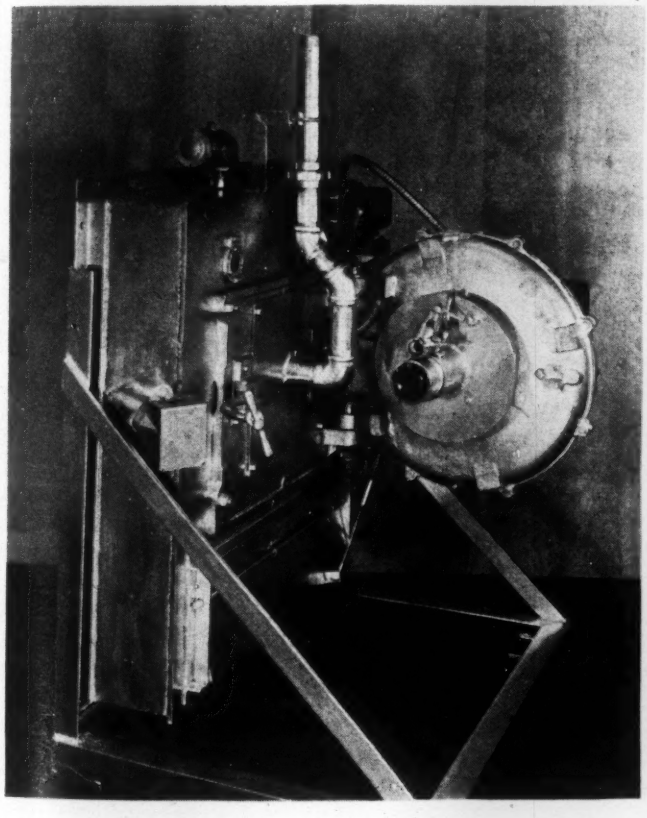
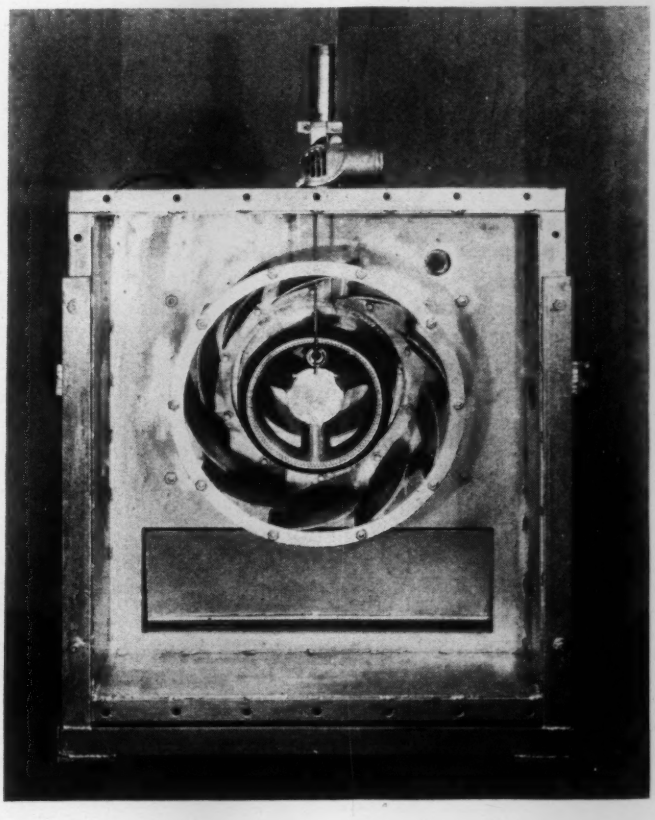


Fig. 2—The burner shown in Fig. 1 from the boiler side. A steel plate is fastened to the boiler, upon which the assembly shown is mounted. The opening in the steel plate has a diameter slightly smaller than the register ring but slightly larger than the gas ring, so that all air entering the burner, firing with either gas or oil, enters concentrically around the burner. This makes for intimate mixing of fuel and air



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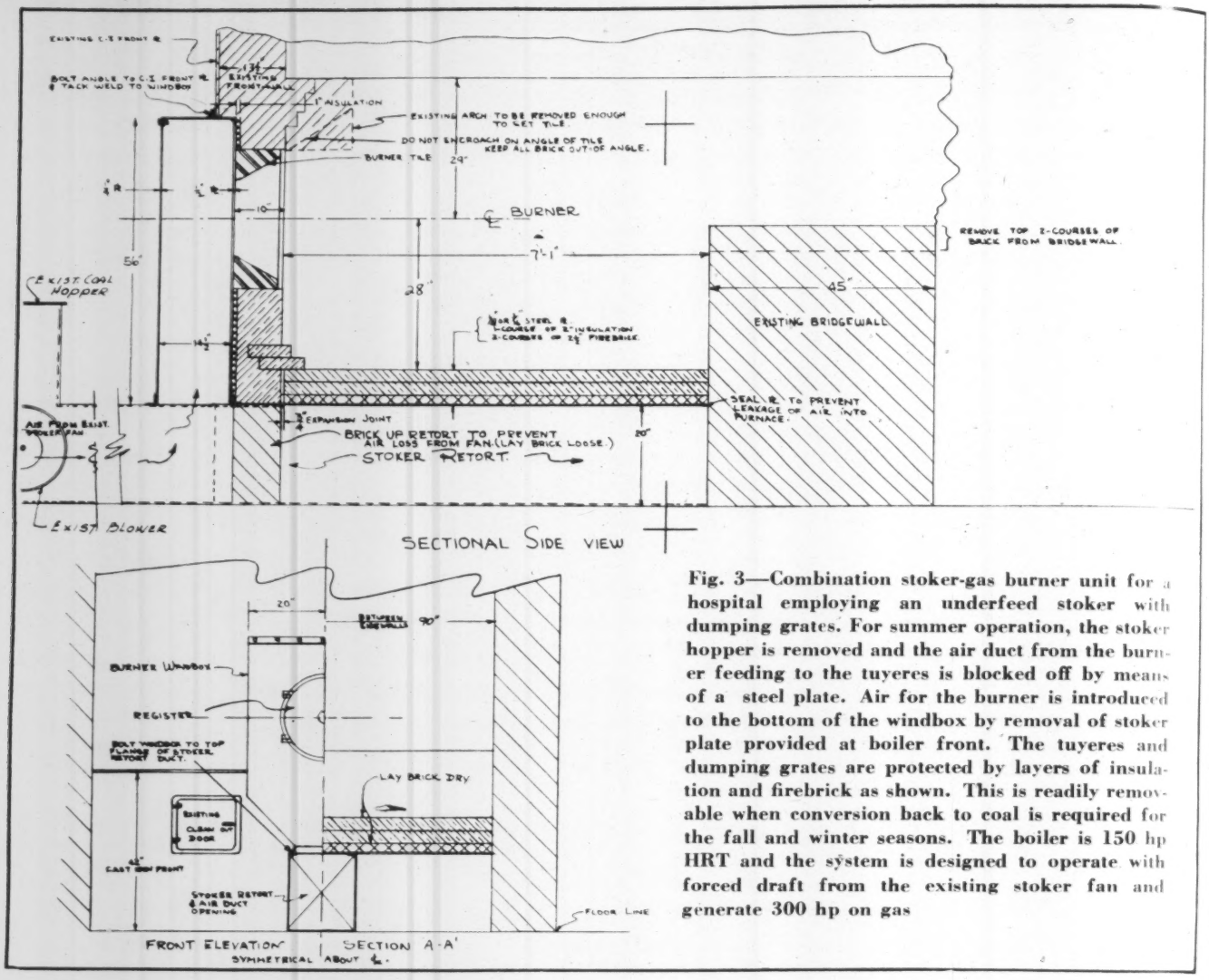

Fig. 3—Combination stoker-gas burner unit for a hospital employing an underfeed stoker with dumping grates. For summer operation, the stoker hopper is removed and the air duct from the burner feeding to the tuyeres is blocked off by means of a steel plate. Air for the burner is introduced to the bottom of the windbox by removal of stoker plate provided at boiler front. The tuyeres and dumping grates are protected by layers of insulation and firebrick as shown. This is readily removable when conversion back to coal is required for the fall and winter seasons. The boiler is 150 hp HRT and the system is designed to operate with forced draft from the existing stoker fan and generate 300 hp on gas

other appurtenances. We must examine the power consumption of the firing equipment and its auxiliaries. Labor costs form part of this study, and these must be determined as accurately as possible to obtain a correct analysis of the problem.

Up to this point, we have been dealing in generalities in the discussion of fuel selection. We have merely stated the problems confronting the designer in selecting this fuel. Let us now examine the step-by-step method by which we may properly arrive at the solution.

The first step is to make an accurate analysis of the steam load. From this analysis, it is possible to select the best type of firing equipment available, commensurate with the load demands. When this has been determined, it is then possible to select a fuel or fuels which might be used advantageously with this type of equipment.

For example, if an underfeed stoker is to be used, the cheapest fuel might be 1¼ in. screenings; if a spreader stoker, a cheaper coal could probably be employed, perhaps ½ in. screenings. If a chain grate stoker were to be used, a still cheaper coal might be advantageous; or, if an oil burner were to be used, the availability and costs of No. 5 and Bunker C fuel oil should be considered. For each type of equipment considered, an overall boiler-burner efficiency should be set up. This, of course, would vary with the type of firing equipment and the type of boiler.

How to Figure Cost of Fuel

Next, we would obtain the cost per 100,000 Btu output for the fuel selected. This cost may be determined in the following way:

Let A equal the cost per unit of fuel, either a ton of coal, gallon of oil, or a unit of gas; B equal Btu per ton, per gallon, or per gas unit [gas may be sold per therm (100,000 Btu) or per 1000 cu ft, depending upon the locality]; E equal overall efficiency of boiler and firing equipment; and C equal cost per therm or 100,000 Btu output. Then

$$C = \frac{A}{(B \times E)/100{,}000}$$

We are now able, by comparing the direct cost of coal with oil or gas, to evaluate a load in terms of the amount of fuel used per month and easily compare the relative costs of each. For example, let us assume that in a 30 day firing period we use 800 tons of 11,500 Btu coal at $10.85 per ton. The overall boiler-stoker efficiency is 64.5 percent. We then find, by using our formula, our cost per therm output to be $10.85 divided by [(23,000,000 × 0.645)/100,000] or $0.073.

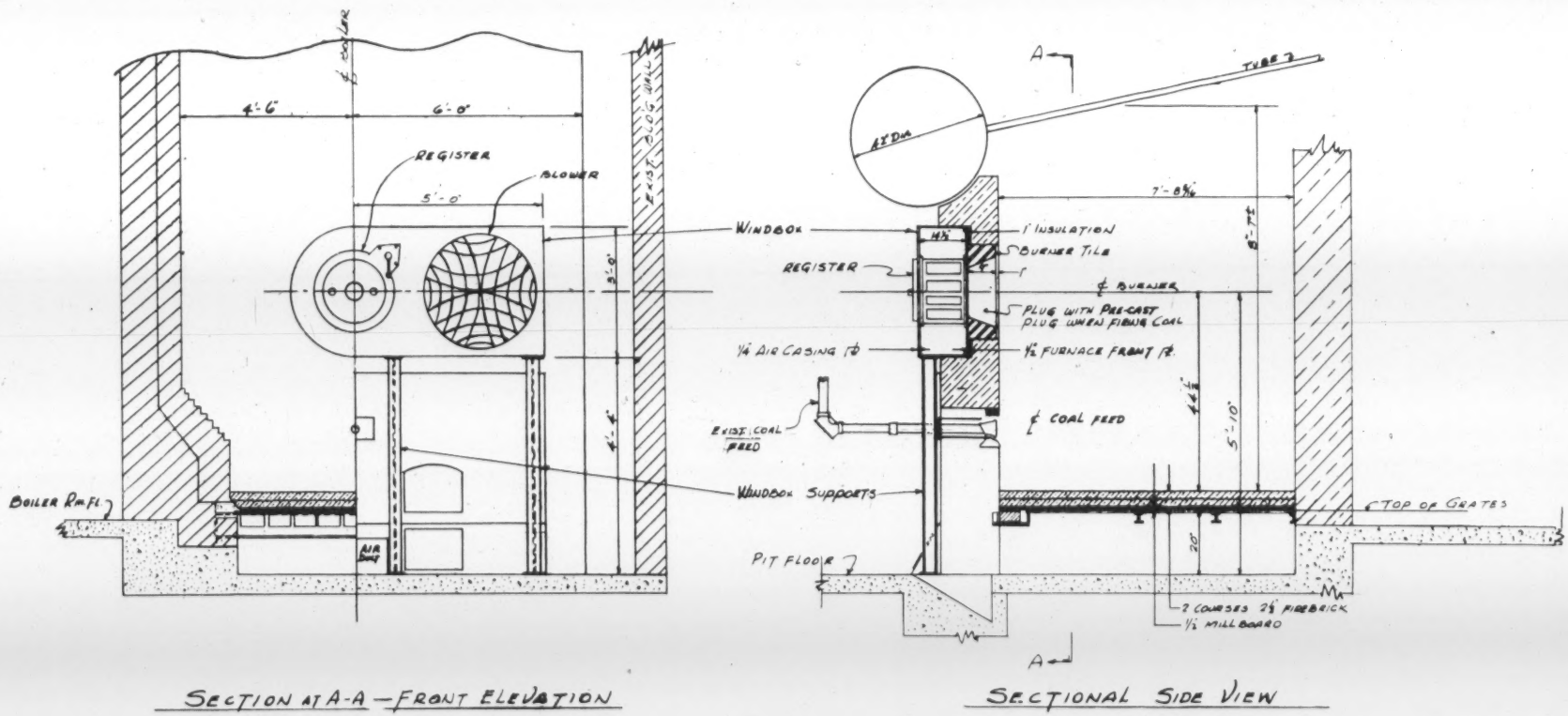

Fig. 4—Gas burner used in conjunction with a spreader stoker. Grates are protected with insulation, and the existing stoker installation is not disturbed. Forced draft is applied to the windbox of the burner by means of a fan mounted directly on the windbox. The boiler is a water tube type

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Let us further assume that a gallon of Bunker C oil having a heat content of 151,000 Btu costs $0.09 and the overall boiler burner efficiency is 74 percent. By use of our formula, we have $0.09 divided by [(151,000 × 0.74)/100,000] or $0.08.

Our coal cost for 800 tons is $8680. The relative cost of oil against coal is $0.08 divided by $0.073. This, multiplied by $8680 gives us our total fuel cost for oil, or $9512.

This completes the first step in our determination of fuel costs—a relative comparison of costs of fuel against fuel only. The next steps involve the determination of fuel handling costs at the plant. In order to determine this, we must calculate the number of firemen employed in coal handling and the time consumed in removal of ash or clinkers. The total amount of fuel handled in tons, assuming coal, divided by these costs, should then be added to the initial per ton cost of the fuel.

To continue with our example, let us say that we employ two men, besides our plant engineer, per shift. The services of one man are required entirely in the handling of coal and ashes on each shift. Let us assume further a cost of handling the disposal of ash. These costs must be added to our initial cost of $8680 to obtain our total fuel cost.

Were we to employ automatic oil firing, it might be possible that we could dispense with the services of one man per shift, due to the fact that we would have no coal or ash handling problem.

Maintenance of Refractory

Our next consideration would be the maintenance of the refractory furnace. It is the author's experience that the maintenance of refractory brickwork in coal fired equipment runs slightly higher than in the case of oil firing; this is, of course, dependent on the method of firing. In plants employing spreader stokers, there is a tendency for the fly ash to fuse in some cases to the side walls. This fused ash is difficult to remove and eventually causes deterioration of the chamber. In the case of chain grate stokers, difficulty is experienced in maintaining the refractory at the grate lines, and in the case of underfeed stokers, difficulty is experienced at the line of the dead plates. In oil firing equipment—generally speaking—if impingement of the flame against the brickwork does not occur, the life of a combustion chamber may be maintained for a longer period of time than for coal firing. In all cases, the assumption is made that the heat release is not excessive.

Another factor in the deterioration of a combustion chamber when burning coal in the midwestern area, would be high sulphur content of many coals which would be offered on the existing market. High sulphur coals produce sulphur dioxide or sulphur trioxide, which attack refractory brickwork. This acid action results in brickwork deterioration and many users of coal therefore insert clauses in specifications as to the maximum sulphur content permissible.

Some fuel oils also have high sulphur content, particularly the Wyoming oils, and fuel oil specifications should be examined carefully to assure that the sulphur content is not unusually high.

The cost of maintaining refractory brickwork must be added to the fuel cost in the cases of coal, oil, or gas as a part of the total operating cost.

Use of Cheap Gas in Summer

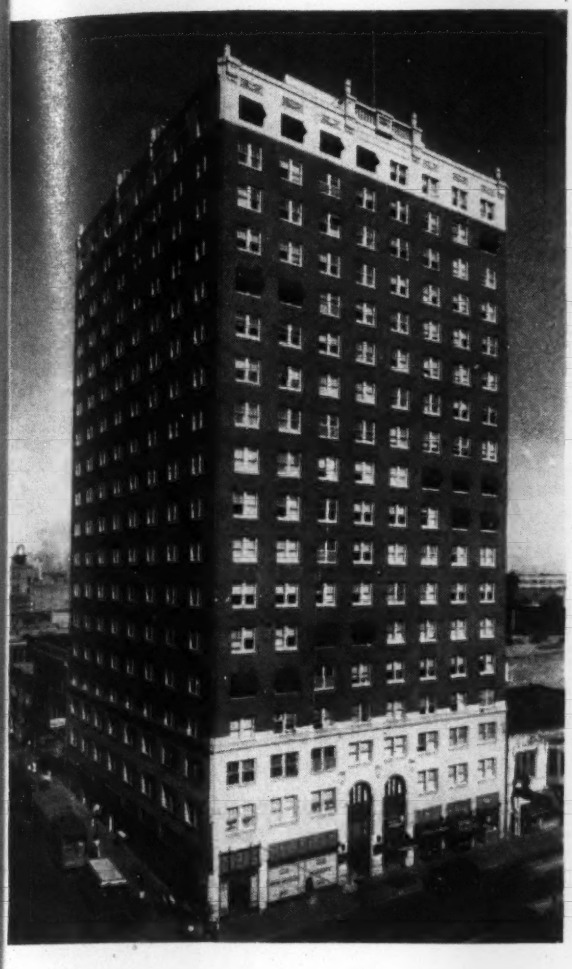
In many localities, natural gas is available and may be purchased cheaply at certain times of the year. This is due to the fact that the gas line serves many consumers for diverse heat and process loads which are considerably higher during the winter months of the year than during the months from April to October. In most cases, these natural gas lines are served from fields in Texas or Louisiana on a contract basis of accepting a specified minimum number of millions of cubic feet of gas per month. Irrespective of whether or not this gas is used, the pipe line company must pay for this minimum. The minimum is determined, in most cases, in terms of the maximum load demanded at peak times, and the ratio of maximum load to average load is called the load factor.

Obviously, if the gas company serves only the same facilities the year 'round, the amount of gas purveyed in the summer would be considerably smaller than the amount of gas handled in the winter. To minimize the difference between the peak and the low load, gas companies have adopted the stratagem of offering relatively cheap gas from April to October, inclusive. In this way, the gas line may be made to handle a relatively high load factor the year 'round.

If this type of summer gas rate is available to the plant being designed, an examination of the rates and the availability of gas mains to the plant should receive careful consideration. In some instances, the gas line may be sufficiently close that no considerable expenditure will be necessitated in order to bring gas into the facility. Applying our formula for basic fuel cost, we may then determine the cost of 100,000 Btu of output for gas, as compared to coal or oil, and if the figure shows that a saving may be effected by the use of gas, we would then study the possibility of the design of the equipment with combination gas and coal or gas and oil. Many arrangements of equipment are possible with combination gas and coal equipment, and some manufacturers design combination gas-oil equipment.

In many localities, interruptible gas contracts are available. These contracts are offered for the entire year, but the user must cease gas firing on demand from the purveyor in a specified number of hours, or the gas dispatcher may give the consumer from day to day a statement as to the amount of gas he may use in a 24 hr period.

It is obvious that this type of contract requires the selection of equipment which must offer the operator a simple means for changing from one fuel to the other. There are at the present time many installations of combination gas with spreader stokers, chain grate stokers, and underfeed stokers; and, in a few instances, there are installations which combine oil with gas and coal in a similar manner. This is particularly true in facilities where gas is offered on an interruptible basis for approximately eight or nine months of the year, with the possibility of but a few days interruption during this time. For these few days, oil is used and for the balance of the year the installation is fired with coal.



30 Days to Condition 18 Story Building

The Irwin-Keasler building, Dallas, is air conditioned with 77 packaged units

AN 18 STORY Dallas office building has just been entirely air conditioned in 30 days and Co-Owner T. K. Irwin gives credit for the accomplishment to labor-contractor cooperation of a type which he thinks "should set a national pattern." The job on the Irwin-Keasler building was done between April 1 and May 1.

Three factors stand out as responsible for the record-breaking installation job, according to Mr. Irwin: willingness of all groups to sit down around a conference table and evolve their agreements, selection of packaged air conditioning units, and the personality of the young structural engineer and former Seabee who served as general superintendent.

A former student at Massachusetts Institute of Technology and a Seabee with a record for action in Attu and Okinawa, General Superintendent G. L. Howell kept his finger on all phases of the mammoth undertaking and helped maintain complete harmony among the 210 workers and six subcontractors involved on the project.

An idea of the magnitude of the project is gained from a few statistics on individual jobs. Sheet-metal workers turned 35,000 lb of sheet aluminum into nearly a mile of air ducts, right on the scene. More than 2000 ft of pipe was installed in the 18 floors of the building to carry the 1150 gpm of water required. Electricians built and installed a giant electrical switchboard to power 159 motors in 10 days. One of the largest cooling towers in Texas was erected.

A million dollars worth of liability insurance was carried on the job, and it is reported that not a single mishap occurred. Tenants suffered but little inconvenience as workmen cut through walls, installed air ducts and water pipes, and boxed in the finished installations.

Seventy-seven packaged air conditioning units were used. These were placed in batteries of 12 on each third floor, to serve the floor on which they stood and the floor above and below.

Matthews Engineering Co., Dallas, furnished the units. This was the largest single order ever received by the manufacturer. Individual units "proved the best type of system, since the building is L-shaped. Too, decentralized units eliminated the need for a large outside conduit entering each floor and blowers to force air horizontally from this conduit." With the basement already fully utilized, there was little room for a large single unit.

The cooling tower, one of the biggest in the state, was erected on a two story portion of the building. It is able to reduce by 10 deg the temperature of the water required.

Work schedules were carefully drawn, timing work so that the task could be completed within 30 working days. So carefully were they worked out that by the third day of actual installation work, Superintendent Howell was able to predict the precise date of completion.

Irwin and Co-Owner T. F. Keasler were so proud of the speed record that they bought full page advertisements in Dallas newspapers on Sunday, May 9, to salute members of the 16 labor unions and contractors and suppliers who had a part in the project.

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Some Problems I've Met in RADIANT HEATING

T. NAPIER ADLAM, vice president of the Sarco Mfg. Co., Bethlehem, Pa., is one of the country's foremost authorities on radiant heating, his experience with it beginning many years ago during its development in England. He gives here his views and opinions on a number of questions that come up in the application of this kind of heating to many types of buildings

PRINCIPAL PURPOSE of this article is to explore some of the problems that apparently confront many designing engineers and architects in deciding on the best way to handle radiant heat installations. Because of the varied nature of these problems it is practically impossible to describe within a reasonable space the many phases of design which seem to present the greatest difficulties. It may be sufficient, however, to mention a few of the more pertinent points which, from time to time, have been brought to my attention, and to analyze them as they appear from various viewpoints, in the hope that some light may be thrown on some of the obstacles encountered by many architects and engineers.

In most cases I find that the major troubles arise from faulty design brought about by inexperience, or by the acceptance of incorrect statements. Some faults arise in connection with the installation work, but these I find are generally confined to air pockets, short circuiting, and similar shortcomings. Here again, these errors invariably are due to lack of experience in this class of work.

As to the actual faults which affect the occupant, it is safe to say that one of the most frequent complaints which reaches me is high floor temperatures, while an equally serious although not as frequent complaint is in connection with drafts. Quite a number of complaints are received in regard to variable room temperatures, but often these occur because balancing fittings have not been installed, or due care has not been given to the regulation of water flow in the various coils after the system has been started.

Quite a number of faults seem to arise because the different types of floor or ceiling construction have not been taken into consideration when the pipe coils were calculated. The question of floor temperatures should receive very serious consideration, especially in view of the many contradictory statements from various sources, many of which appear to have been made without careful analysis. Some of these statements can be classified as bona fide opinions, even though they may appear wide of the mark but, on the other hand, some are so obviously incorrect that I am afraid they can only be classified as the results of poor information on the part of their authors.

Heat Is An Elusive Thing

The difficulty in obtaining reliable data in connection with heating systems is that heat is so elusive. It is often impossible to restrain the action of heat for a sufficient time to make reliable recordings, and since heat is one of those things which will not stay put, it is very easy for those not having a very wide experience with all kinds of heating systems to be mislead when collecting data with radiant heating panels.

When faced with the problem of deciding which position is best for the panels, the engineer will usually find it advisable to settle the issue by examining the floor and ceiling constructions. Often we find the ceiling construction consistent in all rooms, whereas the floor construction may vary considerably from room to room. This difficulty can be met by having two or three zones or by using more or less pipe according to the conductance of the construction or emissivity of the surface. However, if we intend to produce a first class job with comfort in all rooms, we invariably have to use different factors for heat transmission from the pipe coils, since one factor will not apply to all conditions.

I have no particular preference for floor heat over ceiling heat, or vice versa, but a heated floor covered with carpet will give a very mild and pleasant sensation of comfort, providing the surface temperature is not too high. There are many cases, however, where ceiling

heat will not take care of large heat losses at low level and, then again, there are cases where it is practically impossible to use anything but ceiling heat.

Unfortunately, there are many radiant heat enthusiasts who think that radiant heat panels will do anything and everything, regardless of where they are placed. This, of course, is a basic error of the most serious nature.

There are others who think that because radiant heat is essential for true comfort, we should have radiant heat rays to the exclusion of all other methods of applying heat. Obsessed with this idea they recommend that since we obtain heat from the sun which is above us, and because the ceiling panels usually give off less convected heat than those in the walls or floors, the ceiling position is the complete answer for real comfort. This theory also is basically incorrect, although we must admit that some units sold as radiant heat units are practically convectors, since most of the heat given off is convected heat.

Our main object in dealing with any building should be first to insure that we can maintain in each room a mean radiant temperature at all times compatible with the radiation loss from the human body. If we fail to do this we forfeit the true principles of radiant heating, regardless of where we place the panels.

A good heating engineer will first aim to place the panels in the most suitable position to obtain the correct MRT. He will then know from experience whether it is desirable or even essential to supplement the output from the panels with some other method of heating to give comfortable and healthy conditions everywhere.

Incorrect Application Can Cause Drafts

There are cases where floor heat incorrectly applied will create drafts which are found to be most objectionable and difficult to counteract, and, on the other hand, there are instances where ceilings are too high to be very effective. As to the question of drafts, it is an erroneous statement to say that with radiant heating it is impossible to have drafts, and yet this statement has led to many unsatisfactory installations, especially in churches and public buildings.

It is perfectly true that low temperature radiant heating panels will not create convection air currents to the extent we encounter with other types of heat emitting surfaces. But since we do not alter any of the known physical laws by installing radiant heat panels, we shall certainly meet with very objectionable drafts if we create conditions which induce a serious movement of air.

To explain this, I give in Fig. 1 a transverse section of a very beautiful church which, upon request, I visited after the heat had been started and trouble encountered. The floor was heated in the usual way with pipes embedded in the concrete, and the designer had, unfortunately, been lulled into unsuspecting complacency by the usual statement, recorded above, that with radiant heat no drafts would be felt. In this particular case, the church was somewhat high in relation to its width, and several fairly large stained glass clerestory windows were installed at high level. The surface of the center aisle had a rough concrete finish, while the floor surface on both sides of the aisle was covered with asphalt tiles. This area will be used for pews, but because of the draft conditions obtained from the heating system the congregation insisted that work on the pews be suspended until this could be corrected. Chairs were being used in the interim.

Cause of Drafts in a Church

The surface temperatures of the floor under working conditions were found to be fairly consistent, showing that the embedded pipes were distributing the heat evenly, but because of the lower emissivity of the asphalt tiles on the sides as compared with the more effective surface of the rougher concrete aisle, much more heat was given off per square foot of area from the center aisle than from the tiles. Furthermore, still feeling secure in the belief that drafts would not occur where radiant heat was provided, the engineer had evidently ignored the effect of the large clerestory windows and the rapid cooling

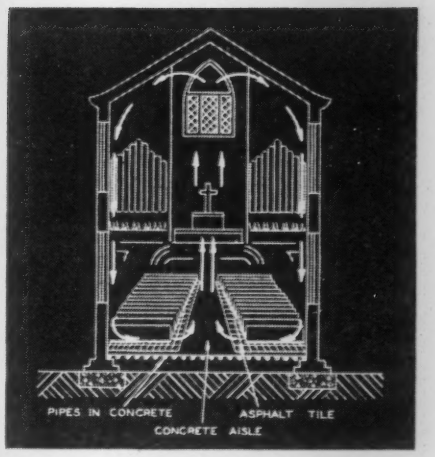

Fig. 1—Air in contact with the center aisle in this church warmed easily and rose rapidly and the cool air descending at the sides was not efficiently heated by the asphalt tile portions of the floor to counteract the draft

effect of the very exposed side walls of the church.

Even with the outside temperature of 40 F which prevailed at the time of my visit, the drafts created, following the course indicated by the arrows in Fig. 1, were most objectionable. The air in contact with the surface of the center aisle warmed easily and rose very rapidly to the roof, as was clearly demonstrated by a cloud test. The descending air was rapidly cooled as it moved over the clerestory windows and side walls and there was not sufficient heat given off from the floor near the outer walls to counteract this falling current of cool air because of the aforementioned inefficiency of the polished asphalt tiles on the sides.

Had the individual heat losses been taken into account and due regard given to the effect of the emission from the different floor surfaces, it would have been a simple matter when erecting the church to have provided some panels, or convectors, in the side walls under the windows, or to have installed some kind of solid baseboard panels along both sides of the church to counteract the effects of drafts. Since the church was already finished before the fault was discovered, the remedy will be a much more expensive undertaking.

Another equally erroneous claim one often hears is that with either floor or ceiling heat it is possible to have open windows without fear of

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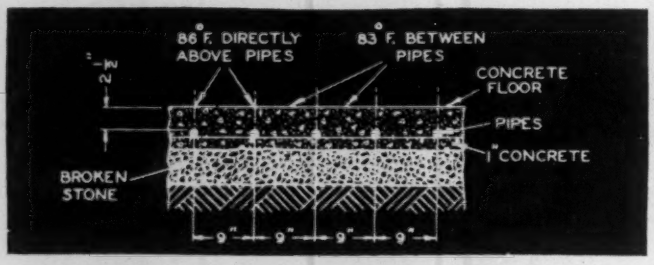

Fig. 2—Surface temperatures which can be expected with the type of floor construction shown and with ¾ in. pipes spaced 9 in. apart and a mean water temperature of 102 F. Surface temperature directly above the pipes is only 3 deg more than between the pipes

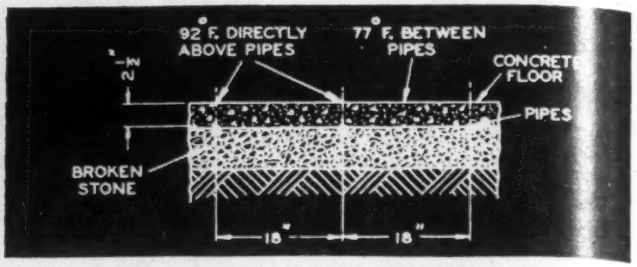

Fig. 3—Surface temperatures which can be expected with the type of floor construction shown and with 1 in. pipes spaced 18 in. apart and a mean water temperature of 120 F. In this case, there may be a difference of 15 deg. as compared with the 3 deg difference indicated in Fig. 2

feeling a draft. While this statement is not literally true, it is possible, of course, to prevent the human body from feeling cold even though the air is at a comparatively low temperature, providing sufficient radiant panel surface is provided to keep the body warm.

It must be remembered, however, that if the human body loses more heat by convection than is compatible with comfort and health, the ratio of heat losses is thrown out of balance and we are not really comfortable. The only safe way to ensure a good radiant heat system is to allow for all the heat losses in our calculations, and then to make sure that the mean radiant temperature of the room is within the limits required by the human body, and that the room air temperature corresponds with the MRT to guarantee a perfect balance of heat.

Heat Distribution Must Be Balanced

More often than not, it does not matter whether we use ceiling heat or floor heat, but it does matter how well the system is balanced to give even distribution and comfort everywhere. We may never be able to locate heating panels in the most efficient manner as theory demands. It behooves us, however, to use common sense and our general experience with heating problems to produce the best results with radiant heat surfaces wherever they are placed.

The cooling effect of large windows must always receive consideration if we are to produce a true comfort effect everywhere. It is not sufficient to place a heated panel above a large window to counteract the cooling effect of the glass. The overhead panel may throw down heat rays theoretically equivalent to the absorbing power of the glass, but the cool air moving down over the unprotected glass of a large window can create a current of air over one side of a person sitting near the window and become very objectionable. Furthermore, it often happens that much more surface of the human body is exposed to the cold windows than is exposed to the panel overhead.

From experiments made with sandwich glass, as used in stratocruiser planes, it seems that at no distant date we shall be able to have our windows electrically heated and thereby rid ourselves of one of the greatest difficulties we are up against in dealing with churches and public buildings.

Returning to the question of floor panels, it seems that much confusion also exists in the minds of architects and engineers as to what can be expected from different types of floor construction and floor surface temperatures, and some very unsatisfactory jobs have resulted in consequence of this confusion. It appears from statements made that many advocates of floor heating believe that because a floor does not burn the feet or feel too warm, everything will be satisfactory. Here again, we are on dangerous ground because we have health as well as comfort to consider.

I have found it to be perfectly true that many people prone to rheumatism and similar maladies derive much benefit from a warm floor but, on the other hand, we also know from experience that persons spending several hours a day with their feet on too warm a surface will be much more susceptible to the effect of cold and wet wintry weather when they go outside. Therefore, it is not correct to assume that any floor temperature which does not actually burn the feet will be equally satisfactory for general health if applied over long periods.

In England, where radiant heating was introduced, it was for many years thought that the surface temperature of the floor should never be more than 70 F to 72 F which, with an accepted room air temperature of 60 F, gives 10 to 12 deg differential between the floor surface and the room air temperature. Since it is the effect of the warm floor on the human body through contact with the feet which is of first importance, the relation between the floor surface temperature and the air temperature appears to be of no real significance. However, this was the first problem that I had to contend with when applying floor heating in America for room air temperatures in the region of 70 F. It was only after several years of careful research that I decided on a safe maximum surface temperature of 85 F. In this connection all my tests were made with floors having very little temperature variation over the whole of the surface, since that is the only correct and satisfactory way of heating a floor.

Proper Spacing of the Pipes

To ensure an even surface temperature it is necessary to calculate the spacing of the embedded pipes somewhat in inverse relation to the conductance of the floor material used above the pipes, and to use pipe of appropriate size to meet these conditions. It is regrettable, in my opinion, to find some advocates of radiant heating recommending and installing large size pipes spaced widely apart, under the pretense that "heat is heat"

however the pipes may be installed.

To show the various results of spacing pipes in a concrete floor, Figs. 2 and 3 give the surface temperatures which can be expected with pipes 2½ in. below the concrete surface, spaced at 9 in. and 18 in. respectively. These temperatures are such that the same amount of heat will be emitted from equal floor area, as would be the case if two identical rooms were provided with different floor panels having the pipes spaced 9 and 18 in. respectively. It will be noted, however, that with pipes spaced 9 in. apart, and water at 102 F, there is only a difference of 3 deg between the surface of the floor directly above the pipe and the surface of the floor between the pipes, whereas with pipes spaced 18 in. apart and water at 120 F there may be a difference of 15 deg.

This latter incorrect phase of radiant heating is not given without my having encountered definite evidence of its effect. A very good example of what can happen when pipes are spaced too far apart was provided by schools which I have visited where both students and teachers were being distracted from their studies because of the resulting unevenness in surface temperatures.

It was reported that, unwittingly, a teacher or student would be standing or sitting with one foot resting directly above an embedded pipe and perhaps one foot on the cooler part of the floor, or with both feet resting above embedded pipes. Since the surface over the pipe covered by the foot could not dissipate heat except by conduction, its intensity at that point very soon became unbearable. The person received a shock and moved his feet hurriedly. The result not only produced a very serious psychological effect on the person concerned, but it was also found to be a continual source of distraction for the whole class. Thus, the work of the entire classroom became actually disorganized by these happenings, as I discovered from my own observations when I visited some of these schools.

Total Heat Emission Affects Spacing

There are times when it is not necessary to space the pipes close together if the total heat emission required from the floor is small. However, in these cases the temperature of the circulating water will also be low and no ill effect will be felt anywhere. If the concrete above the pipes is very thick, or a layer of material having low conductivity (such as a carpet, parquetry, or hardwood top flooring) covers the concrete, the intensity of the heat above the pipes is not so severe because the retarding effect of the medium on the transmission of heat through the material above the pipes will allow more time for the heat to be conducted transversely along the concrete, thus giving a more even surface temperature over the whole of the floor area. It must be remembered, however, that in adding material which retards the flow of heat upward, we are simultaneously making the downward flow of heat proportionally easier and a greater loss of heat is inevitable unless we add considerably to the nonconductive medium below the pipes.

Then again, there is the debatable question of total heat emission from a heated floor, and also from any type of heated panel. With an ordinary floor surface, such as we have with smooth concrete and similarly finished surfaces, it is found that with an average surface temperature of 85 F and the correct conditions of MRT and room air temperature as needed in this country, the total heat transmission which can be relied on is approximately 40 Btu per sq ft of floor surface per hr. This, of course, will vary somewhat if room conditions deviate from standard. For instance, cool air descending from a large window and moving across a heated floor will pick up considerably more heat than stagnant air, providing that the surface temperature of the floor can be maintained. Usually, however, the cool air passing over any portion of the floor will lower the surface temperature locally, so that there is a limit to the extra heat which can be allowed for under such conditions.

Sometimes it is possible to take advantage of this extra cooling effect and utilize border heating near outside windows or outside walls where there is exceptional heat loss. This usually means that adjacent to the windows we place the embedded pipes closer together and, if possible, raise the coils nearer to the floor surface, or we install the warmer flow pipes in that part of the floor under the windows and then continue the coils to the inner and more occupied portion of the room.

Be Careful With Border Heating

Even with these methods of getting extra heat we must exercise care not to overheat any part of the floor which may eventually be used by the occupants, for again we find experience has taught us the necessity of prudence in this respect. I have found at least one instance where border heating had been applied to a large office over an area which was originally expected to be unoccupied, only to find later that it became necessary for some of the office staff to use the floor space around the wall where border heating had been installed. So marked was the effect of the extra warm floor on those occupying this area, and so many were the complaints, that those employees occupying this section of the office were nicknamed "tenderfeet" by the rest of the staff.

Unfortunately, we find that when radiant heating began to reach the competitive field, the factor for heat transmission was boosted to all kinds of fantastic (in my opinion) figures for no apparent reason whatever. First it was raised to 50 Btu per hr, then to 60 Btu, and in some cases where it seems that the imagination had no limit, we find as much as 100 Btu per sq ft per hr mentioned.

Does One Error Balance Out Another?

I believe that, here again, these very high figures, if not simply the results of imagination, can perhaps be attributed to the erroneous conclusions reached by some enthusiasts who used high "U" factors for total heat losses from a building, only to find that in making a balance sheet for their calculations they had some very large heat losses to balance against heat emission. In figuring the heat emitted from a heated floor to balance such calculated heat losses, these engineers would then obtain results wide of the mark of accuracy.

It is claimed by some that by

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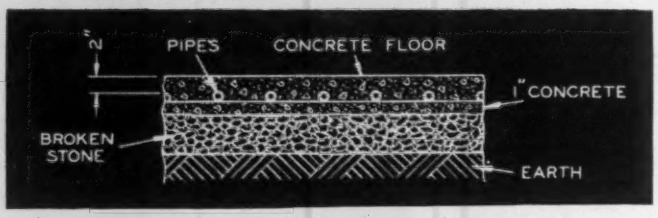


Fig. 4—If some rooms have pipe coils in concrete, as indicated here, the mean water temperature to meet the requirements will be lower than with the constructions shown in Figs. 5a and 5b

using high "U" factors for calculating the total heat losses for radiant heat systems, as is often done with conventional methods of heating, and then employing a high heat emission factor for radiant heat panels, the engineer will find that one error balances the other and everything will be satisfactory. This line of reasoning may satisfy the novice, but it is a very dangerous assumption to make since it can lead to all kinds of errors and difficulties.

It may be true that this method will tend to balance out in some cases; unfortunately, the complete truth may be revealed when it is too late, as on some jobs we may be using correct factors on one side of the equation and incorrect factors on the other, only to discover eventually that we are left with some very unsatisfactory jobs. This invariably occurs when we are dealing with the larger and more important jobs, similar to the churches and schools referred to earlier.

I am fully aware that it is often possible to obtain from fully exposed floor panels a slightly higher transmission of heat than the figures I usually advocate*, but over the years I have found that it is safer to allow for the various contingencies which so often arise, rather than to take chances on something which cannot be corrected without partly demolishing a building. It must be remembered that a large percentage of a floor may be screened by furniture, so that it is very unwise to assume the highest emission factors it is possible to obtain over the whole of the floor area. Then again, while designing a system we may be told by the occupants that no carpets or rugs will be used, only to find that eventually new occupants may want carpets and mats or some other kind of floor covering. If the designer has already used the highest possible factor for Btu emission, there is no margin to meet such contingencies and difficulties begin to arise at once.

It has been suggested—and, within reasonable limits, it is true—that since very low outside temperatures exist for short intervals only, it is possible to allow for a higher floor temperature to take care of these conditions, and to use the more moderate surface temperature range during the average winter conditions. However, it is found that in many cases this suggestion is often carried to the extreme, especially by those who hold the viewpoint that in the event of complaints being registered because of very high floor temperatures, the weather will soon change to a milder condition, allowing a drop in the floor temperature and thus enabling them to elude the consequences. Unfortunately, this quick change in the weather does not always happen as we may anticipate, and in some localities the very low outside temperatures extend over long periods.

Need Margin for Severe Weather

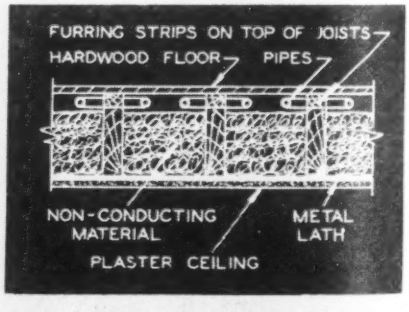
In designing a system we should certainly have some margin to deal with very extreme weather and to cope with unforeseen circumstances, because if we have already banked on the available margin to deal with low outside temperatures which we know will exist, then with radiant heat systems we will have no safe margin for emergenices.

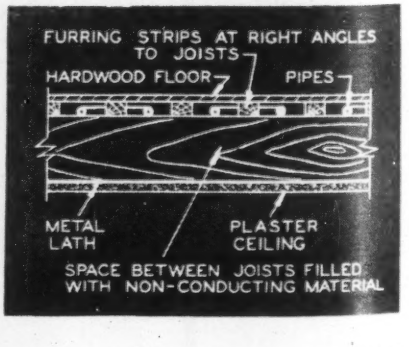
With cast iron radiators and convectors it is generally possible to raise the temperature of the circulating water or the steam pressure to meet any reasonable emergency without creating any serious conditions for the occupants, but when in order to meet conditions which we know will exist at times during the winter it becomes necessary to heat radiant panels to a temperature which is either unsafe for the plaster or unhealthy for the feet, then the heating engineer has failed miserably in his profession.

Another problem which often arises in applying radiant panels to a floor of a building involves the method of construction. If some rooms have the pipe coils in concrete, somewhat as illustrated in Fig. 4, then it will be found that a mean water temperature of about 100 F will meet the required conditions. If there are other rooms with floors constructed with wood joists, somewhat as shown in Figs. 5a and 5b, then the mean water temperature should be much higher to meet the same conditions, probably in the neighborhood of 155 F. Thus it will be seen how impractical it is to use one factor for the heat transmission from a pipe, basing our calculations on the difference between the temperature of the water and the air temperature in the room.

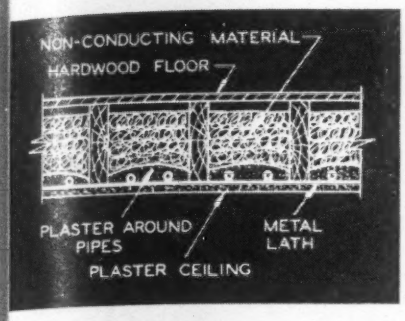
In dealing with these two types of construction on a job, we will find it necessary to use two zones so that one zone can operate at a much higher temperature than the other. Sometimes it becomes necessary to have several zones in order to obtain a correct balance of conditions. Water temperatures will vary, depending on whether the floors are covered with carpets, linoleum, parquetry and so forth, but providing

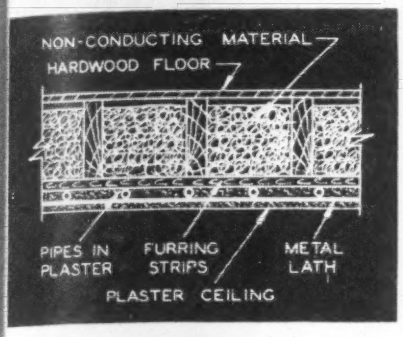
Figs. 5a and 5b—With floor constructions of this kind, the mean water temperature to meet the requirements must be considerably higher than for the application shown in Fig. 4





* See Radiant Heating, by T. Napier Adlam, chapt. 13, pages 228-66.





Figs. 6a and 6b—The author prefers to place the coils above the expanded metal, as shown in these sketches, when dealing with plaster ceiling jobs, rather than to have the coils in the plaster below the metal lath, as shown in Fig. 7

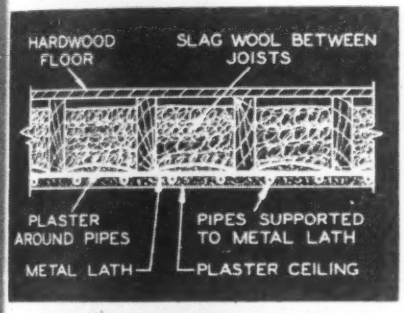


Fig. 7—Pipe coils embedded in plaster below the metal lath. The author prefers the coils above the expanded metal, as shown in Figs. 6a and 6b

the surface of the floor is not too smooth and polished, and providing the conductance of the medium above the pipes is not too great, any suitable floor construction can be adopted.

We find that advocates of floor heating point out the danger of having a cracked plaster ceiling, but floors will also crack if the pipes in the coils are too long or the water temperature too high. Either possibility exists only if precautions are not adopted in designing and carrying out the work.

Because of this danger of cracking I would like to offer a word of caution concerning the use of very long pipes embedded in concrete or plaster without any means of absorbing expansion. We often hear the remark that it is possible to use pipes 50, 75, and 100 ft long embedded in concrete without having any trouble due to expansion. It is not what happens the first year or perhaps the second year; it is what will happen in 10 or 12 years which really concerns the owner.

I have seen installations where a leak developed in the embedded pipes on an average of every month, but it was only after eight years that the trouble really began to show itself. I suggest that these experiences should be a warning to those who think they have learned all about radiant heating in one or two years.

In dealing with plastered ceilings, I prefer to place the pipe coils above the expanded metal, as shown in Figs. 6a and 6b, rather than to have the pipes embedded in the finished plaster below the metal lath, as shown in Fig. 7. With electric conductors and small copper pipes there is perhaps some reasonable claim for having these below the metal lath, although with small copper pipes there is some danger of sagging. Sagging of the pipes will not only form small air pockets, but the greatest danger with sagging pipes is that the thickness of plaster will vary considerably and leave weak spots in the ceiling.

When the pipes are above the expanded metal, the plaster may be applied over the metal lath to an even thickness and will be less weight for the metal lath to support. It is, of course, essential to add plaster above and around the pipes in either case to facilitate the conduction of heat over the whole of the ceiling surface, because if this is not done much heat will be lost and the surface temperature will be uneven. There is also less chance of the surface plaster cracking and darkening with the first method than when the pipes are in the finishing plaster and supported below the metal lath.

Manufacturers of electric conductors for radiant heating recommend that no other metallic reinforcing should be used at any point where it will be crossed by the conductor, because of the hum generated in the wire when the heating element is on.

HOW TO INSULATE VALVES AND FITTINGS

Pipe flanges, valves, and fittings are perhaps the greatest Btu wasters in otherwise efficient power and processing plants. These connections are natural heat wasters for the simple reason that their odd shape makes them difficult to insulate.

However, modern techniques—as evidenced by the mineral wool insulated valves at the new Salisbury Axle plant in Fort Wayne, Ind.—make it a relatively simple matter to effectively insulate irregular surfaces.

The recommended procedure for effectively insulating these surfaces with mineral wool is:

1) Cut and notch blocks of mineral wool to conform to the given contour.

2) Wire the blocks in place with 16 gage galvanized wires or wire bands.

Well-insulated valves save many Btu's

3) Apply a ¼ in. coat of mineral wool cement, troweling it well into cracks between adjacent blocks.

4) Apply finishing: canvas jacket indoors; sheet metal jacket, roofing felt, or asphaltic finish outdoors.

5) For small fittings, insulating cement can be built up in separate coats no greater than ¾ in. thick, to a thickness equal to the insulation on adjacent piping.

Bulletin on Technique Available

Recommended insulation techniques are given in detail in the U. S. Bureau of Standards bulletin CS 117-44 entitled Mineral Wool: Blankets, Blocks, Insulating Cement, and Pipe Insulation for Heated Industrial Equipment. Copies may be had free, in limited quantities, from the Industrial Mineral Wool Institute, 441 Lexington Ave., New York City.

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SUCTION LINE HEAT EXCHANGERS

WHY and HOW They're Used in Refrigeration and Air Conditioning

THE ACTION OF a refrigerant suction line heat exchanger may be readily understood by reference to a pressure-enthalpy diagram of the refrigerant in question. Fig. 1 shows the essential portions of such a diagram including a normal refrigerating cycle.

How Suction Line Heat Exchanger Acts

Beginning with liquid refrigerant at 86 F in liquid line (point "A"), the refrigerant expands through the expansion valve to a lower pressure and temperature (point "B"). Since no heat is added to or removed from the refrigerant during this short time, the conversion to lower pressure will be along the vertical line "AB" at constant enthalpy. The portion "EB" of the horizontal line "EF" represents the portion of the liquid refrigerant evaporated to cool the remainder to the lower temperature. This may be demonstrated by reference to standard thermodynamic tables, the following values of heat being in Btu per lb:

Heat of liquid at point "A"	27.72
Heat of liquid at point "E"	9.32
ΔH	18.40
Heat of vapor at point "F"	78.79
Heat of liquid at point "E"	9.32
Latent heat @ 5 F	69.47

If conversion is at constant enthalpy

Heat of vapor-liquid mixture at point "B"	27.72
ΔH between points "E" and "B"	27.72 — 9.32 = 18.40
Percentage of "flash gas"	18.40/69.47 = 26.5 percent

The refrigerant at point "B," then consists of a mixture composed of 26.5 percent saturated vapor and 73.5 percent liquid. It is to be noted that the heat absorbed in the evaporation of this percentage of the refrigerant exactly equals the heat which must be removed from the liquid to cool it from 86 F to 5 F.

The cycle then passes horizontally along the line "BF" at constant pressure. During this time the refrigerant is absorbing heat in the evaporator, becoming saturated vapor at point "F." At this point, the temperature still being 5 F, the suction line will frost or sweat if immediately taken outside the refrigerated space.

It is necessary, then, to superheat the suction vapor to some point such as "C" in order to obtain a dry suction line. In the absence of a heat exchanger, this superheating can be accomplished either by installing a drier coil (an extra length of suction line in the form of a coil) or by readjusting the expansion valve to obtain a higher coil outlet temperature. If the expansion valve is not readjusted, the refrigerant will leave the coil in a condition represented by point "G" (superheat 10 F, actual temperature 15 F). In any event, the refrigerant vapor must reach point "C" (temperature 65 F) before passing out of the refrigerated space.

H. E. FERRILL, chief engineer of the brass division of the Kerotest Mfg. Co., Pittsburgh, explained the use of refrigerant suction line heat exchangers to increase system efficiency in the April HPAC. He now discusses in detail how such exchangers function and gives practical data on their design and selection

The vapor is then compressed along the line "CD" and cooled and condensed at constant pressure "DA" to complete the cycle.

Referring again to Fig. 1 and standard thermodynamic tables, the wasted cooling effect in Btu per lb between points "G" and "C" may be easily determined.

H @ point "C"	87.35
H @ point "G"	81.23
	6.12

If this cooling effect is utilized in reducing the heat content in the liquid refrigerant, the results are as follows:

H @ point "A"	27.72
ΔH	6.12
H of subcooled liquid	21.60
Final temperature of liquid refrigerant	60 F

Expansion then will be along the line "H-J" and the heat content of the vapor-liquid mixture at the expansion valve outlet will be 21.60 Btu per lb. The percentage of "flash gas" in this case will be (21.60 — 9.32) /69.47, or 17.68 percent. This represents a marked reduction from the 26.5 percent in the previous instance.

Further, if the expansion valve is readjusted to maintain 2 F superheat at the coil outlet, the available cooling effect is increased.

H @ point "C"	87.35
H @ point "G" (S = 2 F)	78.98
ΔH	8.37

Final temperature of liquid refrigerant 46 F.

Heat content of vapor-liquid mixture at expansion valve outlet = 18.35 Btu per lb. (See points "K" and "L").
Percentage of "flash gas" = (18.35 − 9.32)/69.47 = 13 percent.

If this "flash gas" is considered from the standpoint of compressor performance, the importance of keeping it at a minimum can be more readily understood.

Importance of Minimum Flash Gas

In all three of the above cases, the vapor entered the compressor at point "C." Under these conditions the vapor has a volume of 1.703 cu ft per lb. At a temperature of 5 F, the refrigerant has a latent heat content of 69.47 Btu per lb. However, without a heat exchanger, when 26.5 percent of the incoming liquid goes into "flash gas," only 73.5 percent of the liquid is available to perform useful cooling. Accordingly, each pound of refrigerant will absorb only 73.5 percent of 69.47, or 51.00 Btu per lb. The compressor must handle then (1.703/51.00) × 1000 or 33.40 cu ft of vapor for each 1000 Btu absorbed in the evaporator.

Tabulated for ease of comparison, the three cases are:

Case I—No heat exchanger. Percent "flash gas," 26.50. Cu ft per 1000 Btu cooling effect, 33.40.
Case II—Heat exchanger. Superheat, 10 F. Percent "flash gas," 17.68. Cu ft per 1000 Btu cooling effect, 29.80.
Case III—Heat exchanger. Superheat, 2 F. Percent "flash gas," 13.00. Cu ft per 1000 Btu cooling effect, 28.20.

Cases II and III, then, represent an actual increase in compressor capacity over Case I of 12 percent and 17 percent respectively. Such improvements in the operating characteristics of any system certainly should not be overlooked, particularly since no other part of the system is penalized to effect the improvement. The only case where this reasoning does not hold true is that of an insulated suction line, and these are few and far between with "Freon 12" systems.

Subcooling Liquid Cuts Compressor Load

It can be seen from the above analysis that the proper application of suction line heat exchangers for the purpose of subcooling the liquid refrigerant greatly reduces the load on the compressor and decreases its running time and operating expense in the same proportion. In many borderline cases, a heat exchanger can be the deciding factor between a successful installation and one which is a failure. This function of the heat exchanger is its most important.

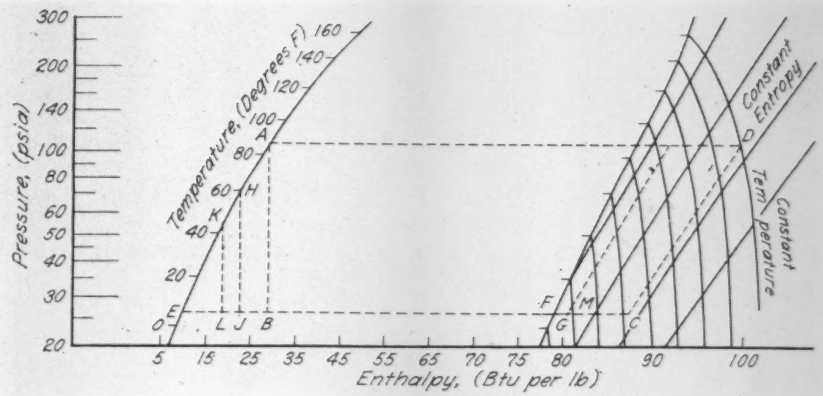

Fig. 1—Pressure-enthalpy diagram for the refrigerant dichlorodifluoromethane. This diagram is for the purpose of illustrating the text. Actual values used in the computations in the text were taken from thermodynamic tables

However, the device—when properly used—also has an important effect on coil performance and capacity. This can be readily understood by a consideration of the following.

Heat Exchanger Aids Coil Performance

With a thermostatic expansion valve operating at 10 F superheat, most of the coil is filled with a liquid refrigerant and saturated vapor mixture. This mixture is largely liquid at the inlet end of the coil, the proportion of vapor increasing as it moves through the coil and absorbs heat. Somewhere near the outlet end, the refrigerant becomes 100 percent saturated vapor, then continues to absorb heat until it reaches a condition of 10 F superheat at the expansion valve bulb.

Referring to Case II above, and Fig. 1, the refrigerant is 82.3 percent liquid at the expansion valve. During passage through the coil and complete evaporation, each pound of refrigerant will absorb 82.3 × 69.47 = 57.1 Btu. After reaching saturation each pound will absorb only 2.44 Btu (difference of enthalpy between points "F" and "G"). A comparison between these two heat values is indicative of the proportions of the coil utilized for the two portions of the conversion from 82.3 percent liquid to 10 F superheated vapor.

In other words, 2.44/82.3 or about one-fortieth of the coil surface is used to superheat the vapor. In the balance of the coil, the refrigerant will remain at a constant temperature of 5 F (disregarding pressure drop), while in the one-fortieth of its length, the refrigerant will increase in temperature to 15 F. Obviously, this outlet portion is not as efficient as the balance of the coil in the removal of heat from the refrigerated space. This factor is considered in good coil design, however, and consequently almost never causes difficulty.

On the other hand, in some cases the layout of the installation is such that it is necessary to lead the suction line out of the refrigerated space almost immediately beyond the point of attachment of the expansion valve bulb. Hence, it is sometimes necessary to readjust the expansion valve, arriving at a higher coil outlet temperature in order to provide a dry suction line. The superheat in the vapor at the coil outlet will be increased accordingly.

If this superheat can be assumed to be a definite value, the effect on the coil performance can be ascertained. If the coil outlet temperature after readjustment is taken to be 40 F rather than 15 F, the superheat is correspondingly increased from 10 F to 35 F. The amount of heat absorbed by the refrigerant vapor in attaining this superheat is 4.93 Btu per lb (difference in enthalpy between points "M" and "F" on Fig. 1).

Comparison of this value with the total heat absorbed per pound of refrigerant shows that the portion of the coil utilized for superheating the vapor has been about doubled to one-twentieth. This increased coil area operating at a higher temperature means that the balance of

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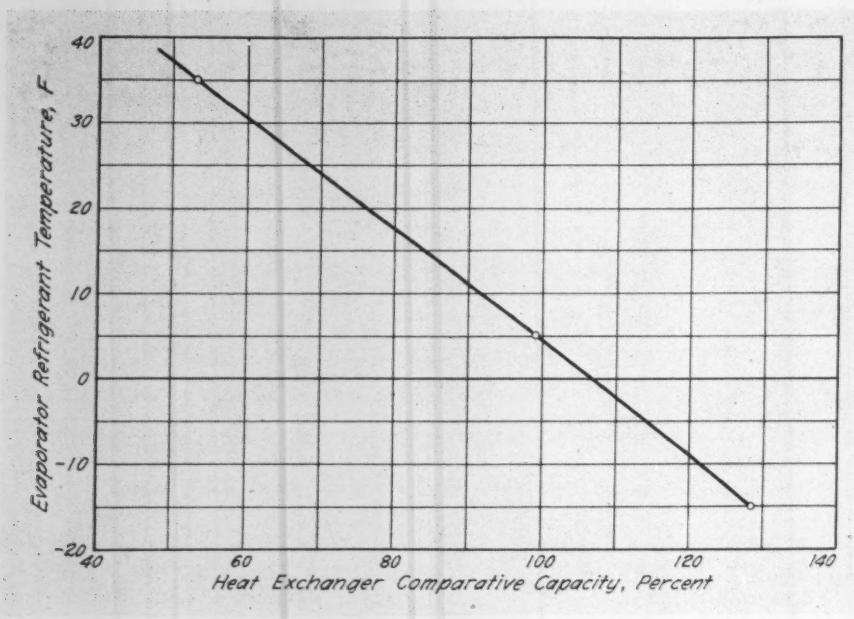


Fig. 2—Suction heat exchanger capacity vs. evaporator refrigerant temperature

the coil must operate at a lower temperature in order to handle the designed load. Lower coil temperature means lowered suction pressure at the compressor, with increased running time due to reduced capacity.

Naturally, the reverse is true if the expansion valve is readjusted to 2 F superheat after the installation of a heat exchanger. The portion of the coil used to superheat the vapor becomes negligible, thereby increasing the effective cooling surface to practically the entire area of the coil.

This equalization of coil temperatures which can be gained by the proper application of heat exchangers may be utilized to advantage in many ways. In medium temperature applications, it insures uniform and complete defrosting during off cycles. In air conditioning applications, it renders the control of humidity more exact since the coil temperature has the most direct influence on the humidity. Of course, it can also be the final influencing factor in the correct functioning of the installation in borderline cases.

Design of Suction Line Heat Exchangers

From the above it can be seen that the overall efficiency of the installation can be increased considerably by the transfer of relatively small amounts of heat from the liquid refrigerant to the suction vapor. Since the available cooling effect is small, it is of fundamental importance that the device used for effecting the heat transfer be as efficient as possible. Since the suction vapor is moving at a rather high velocity, the design should make some provision to allow sufficient time for the transfer to take place. An efficient method of doing this is to enclose the suction line within the liquid line, or vice versa, for their entire length. However, either method is expensive, the former being more so due to larger tubing sizes involved, and neither affects the transfer soon enough to be entirely contained within the refrigerated space. Any other method involves the slowing down of the suction vapor by inducing increased turbulence in its flow. This is the most common procedure but must not be carried to extremes. Any slowing down of the suction vapor increases the pressure drop between the coil outlet and the compressor and therefore must be kept at a reasonable value. About 3 to 5 psi pressure drop through the suction side is customary.

The pressure drop in the liquid line being less critical, it is common practice to extend its path considerably. Both liquid and vapor sides should contain considerable extended surface in order that the device as a whole have as high a transfer coefficient as possible.

In order to combine the features mentioned above with economy of manufacture, heat exchangers have been built in various types. Recent designs have featured compactness and neatness of installation and several efficient models have been produced. They incorporate considerable extended surface on both liquid and vapor sides and induce a certain amount of turbulence in the suction vapor in order to increase the heat transfer efficiency.

In this connection, it is interesting to consider the relative locations of the two sides of the device. Better results will be obtained if the liquid is placed outside the vapor. It will be remembered that the suction vapor does not possess enough cooling capacity to completely subcool the liquid. Accordingly, since the heat exchanger is installed within the refrigerated space, additional cooling effect can be obtained from the surrounding air. The final temperature of the liquid can be reduced to a lower point if it is exposed to this additional cooling effect.

Selection of a Heat Exchanger

The selection of a heat exchanger for any specific application should follow the above general recommendations as to type, design, and location. However, it must be remembered that the capacity of such a device is a function of the temperature difference existing between the liquid and vapor sides. Hence, a heat exchanger of a capacity entirely adequate for a 1 ton coil operating at a temperature of 10 F would be undersized if applied to a 1 ton air conditioning coil operating at 35 F.

The capacity of a heat transfer device is determined from the formula:

$$Q = AU \times [\theta_a - \theta_b]/[\log_e(\theta_a \; \theta_b]$$

where

Q = capacity in Btu per hr, A = area of transfer surface, U = overall transfer coefficient, Θ_a = initial temperature difference, and Θ_b = final temperature difference.

The last expression in this formula is known as the logarithmic mean temperature difference. In any one heat exchanger, the other factors being constant, the capacity then is a direct function of this logarithmic mean temperature difference ($\log \Delta t$).

If we assume three different evaporator temperatures, the liquid and

final suction temperatures remaining constant, we may evaluate these log Δt's for each condition as a basis of comparison. In each instance 100 percent efficiency of heat transfer is assumed and the final liquid temperature is obtained by consideration of the amount of heat removed from the liquid by the suction vapor. Initial liquid temperature is assumed as 86 F and final suction line temperature as 65 F in all cases.

Coil Temp.	Vapor Inlet Temp.	Liquid Outlet Temp.	Log Δt
−15 F	−13 F	39 F	54.6 F
5 F	7 F	46 F	42.1 F
35 F	37 F	73 F	22.6 F

Graphically represented, this relation appears as in Fig. 2. Obviously since the log Δt is the only variable, the bottom of the chart may be labeled "comparative capacity."

Relatively speaking, then, it can be easily seen that if the capacity of a suction line heat exchanger is 100 percent at a 5 F evaporator temperature, the capacity at 35 F is approximately 55 percent and at −10 F, 120 percent. This curve represents, of course, the conditions under a constant entering liquid temperature of 86 F, which is accurate enough for most instances. A higher entering liquid temperature due to a higher temperature of the condensing medium would naturally increase the mean temperature difference and hence the capacity of the heat exchanger. Conversely, a lower liquid temperature would lower the capacity.

While the foregoing computations were made on a purely theoretical basis, the arguments presented are entirely from the practical standpoint. In practice, complete utilization of all the available cooling effect is exceedingly difficult to attain. Accordingly, the liquid refrigerant will not reach as low a temperature, and the "flash gas" will be correspondingly higher than the values presented. However, even though the represented conditions are impractical to achieve in their entirety, the results accomplished are proportional to the engineering skill and effort applied in the design and application of heat exchangers. Any of those offered for sale by reputable manufacturers are well worth their price; and any designed and built on the job to fit a specific installation should adhere as closely as possible to these design and application requirements.

Welded Steel Barge Stores Standby Oil

To be prepared for any fuel shortages that would result in slowing or stopping glass production at its Creighton, Pa., plant, the Pittsburgh Plate Glass Co. recently completed installation of standby oil-burning equipment. The burners, supplying heat for manufacturing glass, now have three sources of fuel—gas from company-owned wells, gas from a public utility, and fuel oil.

One of the major problems was finding adequate storage facilities for bunker C oil. No space was available on the plant grounds to build a tank of sufficient capacity.

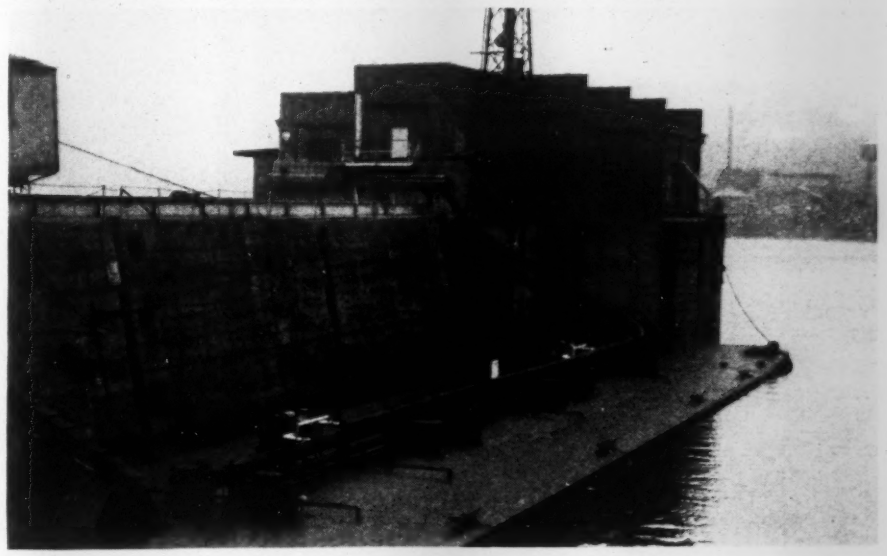
As the plant is located along the Allegheny river and has harbor facilities for unloading sand from barges, it was decided that the solution to the problem would be storing the oil in a barge. This also would answer the question of transporting oil to the glass plant. The barge could be towed to oil company docks, loaded, and returned, with a resultant substantial saving in transportation costs.

A standard 195 ft welded steel oil barge was selected for this service The barge was already under construction as a stock unit, so prompt availability was possible. Since bunker C oil, at low temperatures, has the consistency of tar and will not flow freely, it was necessary to make substantial changes to adapt the barge for cold weather storage of the oil.

Steam heating coils, in a sinuous pattern, are installed in the bottoms of the eight cargo tanks. Steam is supplied from the plant through a 4 in. main. The 3 in. oil discharge line on the barge is protected by a 1 in. steam chaser. On deck, the steam and condensate lines are in one conduit, the oil cargo and steam chaser lines are in another. Both conduits are encased in a larger one. Steam coils also are placed around the suction for each cargo tank. The existing 8 in. cargo discharge line, standard on the oil barge, is blanked off as it is used only during loading at the refinery. A bypass is provided on shore to permit loading oil from tank cars to the barge through the regular 3 in. cargo line. Regular size oil pumps normally used on this type barge would be much too large for the rate of discharge required for this service. Instead, two piston type 37 gpm electric pumps, with 550 volt motors, are employed. The pumps are kept warm by steam chaser lines, and are weatherproof and explosion-proof. Either pump can take suction from any of the eight cargo tanks.

With a capacity of approximately 350,000 gal of bunker C, the barge holds almost a full year's emergency fuel requirements.

Floating fuel oil storage tank is a welded steel oil barge moored to a concrete pier by specially designed timber spars connected in a pivot arrangement that permits the barge to ride high and low water. (Courtesy Dravo Corp.)



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INDUSTRIAL AIR CONDITIONING

F. F. STEVENSON, designing engineer, presents here a summary of factors he considers when choosing the proper air conditioning method for an industrial building. Air conditioning has become a major mechanical feature of all modern structures . . . . . .

SELECTION OF THE right air conditioning system for a building depends upon an engineering analysis of the particular conditions affecting the project. It is thus impossible to generalize or lay down a rule or set of rules to govern the choice of any of the various methods that may be employed.

In his own engineering work, the author has found that a "check list" of the factors to be considered is useful in assuring that no pertinent points will be overlooked in making the correct choice. In setting down these various items, it must be remembered that some of them may not necessarily apply at all under certain conditions, and that under any circumstances some of the points mentioned will be outweighed by other considerations and will therefore not apply. The factors that the author checks are summarized in this article.

It was pointed out in the May issue that there are, generally speaking, two basic methods for cooling and dehumidification. The first method is to cool and dehumidify the air in a simultaneous process, so to speak, by reducing its temperature so that its leaving dew point is low enough to handle the latent heat load. The second method is to dehydrate the air to the required point by one of several solid or liquid adsorbent or absorbent processes and to sensibly cool it to the required degree.

Check factors for using well water, mechanical cooling, and chemical dehumidification methods were given in the May article. Mechanical cooling is considered in more detail this month, and later articles will discuss coordination of the architectural and mechanical designs for industrial buildings.

A more detailed comparison between dehumidification by means of lowering the dew point temperature of the air or through an adsorbent or absorbent process may be apropos at this point.

With the first method, the air must be cooled to a certain point in order to condense out the moisture. If considerable amounts of moisture are to be removed, the air is too cold to use and must be reheated. This is a two way loss in the efficiency of the cycle. Also, it should be pointed out that surface cooling coils cannot be used if *close* control of the dehumidification is required. Close control can only be achieved by an air washer or sprayed coil delivering saturated air at the specified temperature.

On the other hand, the dehydrators using adsorbents or absorbents deliver air at too warm a temperature to use and aftercoolers are necessary. This is especially true of those using silica gel or activated alumina. With the lithium chloride dehydrators, reheat may be needed in cold weather. As stated in the previous article, if supply air with a moisture content of less than 40 to 45 grains per lb is required, then the dehydrators are favored, in the author's experience. Above 45 grains the cooling method is more often used, with a washer necessary for close humidity control, in the author's opinion.

If method No. 1 is used, there is a choice between a multiple unit system—usually several reciprocating compressors, with water condensers and direct expansion coils located in the areas they serve—or a central station system which may be a large reciprocating compressor, connected to direct expansion coils or to a water chiller, a centrifugal refrigeration machine connected to a water chiller, or a steam jet refrigeration machine chilling water. There are a number of variations of these choices, but we shall confine ourselves here to the so-called multiple unit system and a central station system using chilled water. The selection involves a number of controversial points upon which much has been said and written.

In making a choice between multiple direct expansion units as compared to a central station system, it is the author's practice to check the following items:

Initial cost of each type of installation. In installations of moderate tonnage, motor driven reciprocating or radial compressors, with condensers, serving direct expansion cooling coils are usually economical in respect to first cost. In the higher tonnage field, high speed centrifugal machines or steam jet ejectors are favored, with steam jet machines being the lower priced of the two.

Cost of operation. This depends somewhat on use. With multiple units, some may be completely shut down on low loads with commensurate savings. With central station systems, some types have a good operating efficiency at low loads. If steam jet machines are used, water requirements are greater.

Maintenance cost of each type of installation, which is affected to some extent by location of equipment at several points as compared to a central location.

Bulk and weight of equipment, which is decentralized in one case and concentrated in the other.

Space required. Total is greater with multiple unit system, but in some structures the use of a number of smaller spaces with lighter loading is of advantage. With central system, total space may be less but heavy loading must be considered.

Installation must be considered from the viewpoint of the available suitable locations in the structure. Water pumps and considerable piping required for central system using chilled water.

Depreciation rates for the two types of installations.

Performance. Multiple unit sys-

tems offer good flexibility and involve one heat transfer. Central station systems involve two heat transfers and their flexibility may be limited.

Speaking generally, as the tonnage of the installation increases, the choice tends to swing more toward the central station system, using chilled water.

With the central station system using chilled water, the cost of water piping is considerable. However, if cooling coils are strategically located it will be possible to save in air duct construction. With the multiple direct expansion system, we have the problem of refrigerant piping and pressure drop but the run of piping is much less. It may or may not be possible to save in the air duct work. Remember that the running of covered pipes and insulated ducts are two of the most expensive items of present day mechanical construction and any reduction in their length or size will effect large savings.

The Refrigerant Condensing Medium

With mechanical cooling, decisions must be made with regard to the refrigerant condensing medium. Shall water cooled condensers or evaporative condensers, for example, be employed? With steam jet systems, water cooled surface or barometric condensers are used with water recirculated in former and wasted in latter. With centrifugal systems, water cooled surface condensers are used. With reciprocating machines a choice must be made between water cooled surface condensers and evaporative condensers.

Check initial cost of surface condenser and cooling water as compared to evaporative condenser. The latter may have the advantage.

Check cost of operation, with consideration to possible higher power costs in operating refrigeration machine with higher head pressures common to evaporative condenser operation.

Check cost of installation, giving consideration to the weight and bulk of the tower, with required accessories and the duct work requirements of the evaporative condenser.

Check ease of installation, giving consideration to possible roof reinforcement needed for tower, its weight and bulk, and the possible duct work requirements of an evaporative condenser.

Check maintenance cost, with consideration to necessity of frequent cleaning of evaporative condenser and possible replacement of duct work carrying moisture laden air.

Check depreciation rate with each type of equipment, with consideration to the scaling often prevalent in evaporative condensers.

Check performance of each type of equipment, including head pressure against which the refrigeration machine operates.

Check the make-up water required with cooling tower and evaporative condenser.

Check pumping requirements for each type of equipment. Here the evaporative condenser may have the advantage.

Surface condensers are used with centrifugal compressors and decisions regarding condenser water should be predicated on the following considerations.

Can water that has previously been used in manufacturing processes be used for condensing service?

Can condenser water be re-used after leaving the condenser for manufacturing processes?

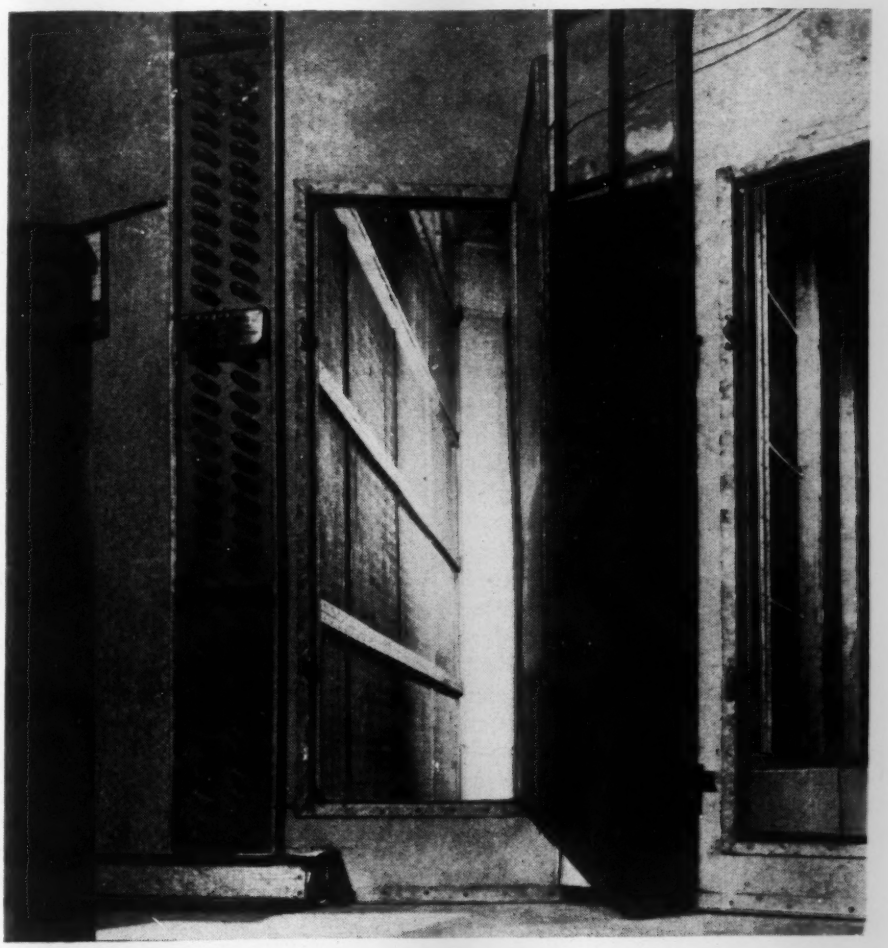
Can the condenser water be wasted?

If the answer to any of the above questions is "yes," there is no need to recirculate the condenser water. If the condenser water must be recirculated, consideration must be given to the means of cooling this water.

Atmospheric and Mechanical Draft Cooling Towers

With surface condensers and recirculated water, a choice must be made between atmospheric and mechanical draft cooling towers (except that atmospheric towers are seldom used with steam jet refrigerating machines because of the possibility that during a hot day with little breeze, the temperature of the water off the tower would be so high, that the condensing pressure of the steam jet machine would go above the critical point and refrigeration would cease.)

A bank of four 24 by 120 four row direct expansion coils utilizing "Freon"



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Check initial cost of each type of tower. Here the atmospheric tower may have the advantage.

Check cost of operation of each.

Check installation costs, which of course vary with bulk and weight. Here the mechanical draft tower may have the advantage.

Check maintenance costs. The mechanical draft tower will probably require more maintenance.

Check depreciation rates.

Check bulk and weight of each type. Here the mechanical draft tower has the advantage.

Check performance of each type, remembering that the performance of an atmospheric tower is dependent on wind conditions as well as wet bulb temperature.

Check water pumping head with each type.

Check amount of make-up water required in each case.

Types of Refrigeration Machines

Using chilled water as the cooling medium, there are three choices of refrigerating machines; steam jet, centrifugal, and reciprocating.

Steam Jet Machines

Check initial cost of steam jet machine, with either surface or barometric condenser, against centrifugal or reciprocating machines using surface condensers. Cost of a larger mechanical draft cooling tower must be also considered with a steam jet machine because of the greater water requirements for steam jet refrigeration.

Check cost of installation, the increased cost of installing a larger cooling tower being a factor with the steam jet machine.

Check cost of operation—the cost of steam and the cost of water, of course, being the factors involved. The steam pressure is also a consideration. Steam jet machine will not show too great power cost increases with higher temperature water. However, if water temperature exceeds the maximum design point, the steam jet machine will not refrigerate at all. Steam jet machine will show good efficiency with reduced loading through the shutting off of one or more of the steam boosters.

Check space required by steam jet machine, including space for larger tower.

Check bulk and weight of steam jet machine, including the tower.

Check make-up water required.

Check control characteristics.

Centrifugal Machines

Check initial cost of centrifugal machine, including cooling tower, using either electric motor or steam turbine drive. Cost of centrifugal machine may be higher than steam jet or reciprocating.

Check installation cost.

Check cost of operation with steam turbine drive or electric motor, which depends on cost and pressure of steam and cost of electric current.

Check control characteristics and efficiency at low loads. The steam turbine driven centrifugal machine has good efficiency at reduced loads.

Reciprocating Machines

Check initial cost. The system may be a multiple one, instead of being one large unit, the multiplexing of course affecting the cost.

Check space and loading requirements, which will vary with the multiplexing. In some building designs, it is easier to find a number of locations to accommodate less bulk and weight per unit than to provide one location for greater bulk and weight.

Check efficiency at reduced loads. The capacity reducing mechanisms built into the larger reciprocating machines increase the reduced load efficiency. Two speed motors do not give great efficiency with light loads.

Check use with evaporative condensers. This may be advantageous in various respects.

Check control characteristics, which may not be as good as with steam jet or centrifugal machines.

Check cost of operation. With a well designed multiplexed system, it may be possible to shut off some units entirely under certain conditions, thus effecting substantial savings when loading is low.

Speaking generally, with loads of 150 tons and under, the reciprocating system deserves consideration, in the author's experience.

Chilled Water Coils and Washers

In addition to the choice of the refrigerating method, the proper selection between chilled water coils, direct expansion coils, and washers must be made. Following are some of the items it is the author's practice to check when deciding between mechanical cooling with chilled water coils and mechanical cooling with a washer using chilled water.

Check initial cost of coils (remembering that the cleanable type must be used with water) and washer.

Check installation cost. Here the coil may have the advantage.

Check depreciation rate, allowing for depreciation of eliminators and spray nozzles with washer.

Check maintenance, remembering that coils may get slimy and need cleaning, and that washer nozzles will clog and eliminators may require frequent replacement if air contains much smoke.

Check operating costs. Here the coil may have the advantage.

Check space requirements of coils and washer. Here the coil has a decided advantage.

Check water requirements, which depend on temperature rise of water with coils and number of spray banks used with washer. A washer usually requires considerably more water for a given load than a coil.

Check pump and pump power requirements.

Check on water recirculation. Water may pick up dirt and sludge in washer sump.

Check air filters required. Water sprays will not eliminate smoke and soot.

Check on mean temperature difference and its effect with each type of equipment. Here the counterflow coil has an advantage.

Check on control of humidity. Coil usually delivers unsaturated air, but does more dehumidification than washer. If washer delivers saturated air, control of humidity can be excellent, which is especially advantageous where narrow limits of relative humidity are required.

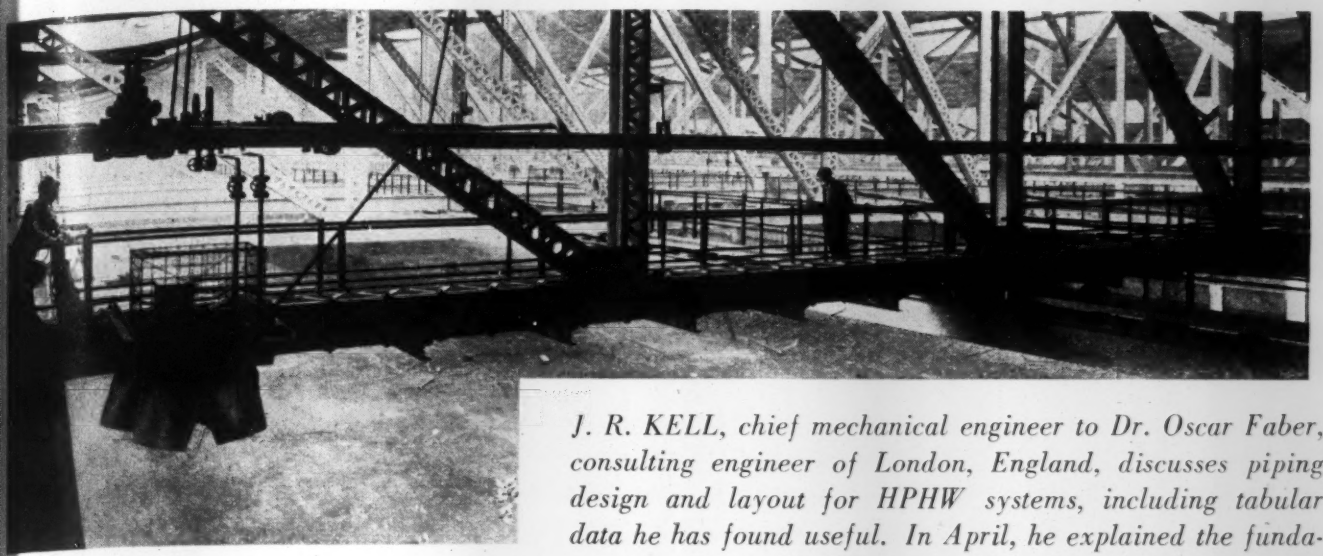
Direct Expansion Coils and Washers

In comparing mechanical cooling with direct expansion coils and with a washer using chilled water, the following items are checked.

Check initial cost of each type of equipment. The direct expansion coil may have the advantage.

Check installation cost. Here the coil has the advantage.

[*Concluded on page 106*]



This downflow unit heater in a large aircraft plant is served by HPHW

J. R. KELL, chief mechanical engineer to Dr. Oscar Faber, consulting engineer of London, England, discusses piping design and layout for HPHW systems, including tabular data he has found useful. In April, he explained the fundamentals of using HPHW for process and space heating; future articles will cover boiler installation and general design

HIGH PRESSURE HOT WATER

Pipe Sizing, Design, and Layout

In the case of high pressure hot water systems, temperature drops up to 100 F may be used, owing to the high initial temperature of the water.

If process heating or a steaming calorifier (heat exchanger) or similar usage is served from the system, the temperature drop between flow and return of the primary water must be such that, while the return is above the temperature of the secondary system, the flow temperature does not call for too high a pressure.

If, for instance, the secondary system is required to work at 250 F the lowest primary return temperature permissible might be about 270 F, and if the temperature drop between the flow and return were 100 F, the flow temperature would be required at 370 F, equal to a saturation pressure of 159 psi. If the primary system were designed for a temperature drop of 50 F instead of 100 F, the flow temperature would then be 320 F, corresponding to a saturation pressure of 75 psi.

Where straight space heating is to be provided without process usage, a temperature drop of 100 F appears to be a satisfactory basis of design, and is one commonly adopted.

What Determines the Pressure?

The pressure at which the system is to work is thus determined by: (a) Whether there is a process requirement, and if so, at what temperature it works. (b) The desirable temperature for the heat emitting elements, where straight space heating only occurs.

There is no point in attempting to economize in the area of heating surfaces by pushing up the pressure and temperature, if the latter means using heavier piping and flanges and heavier boiler construction. Allowing for the 100 F drop, and a return temperature of the order of 220 to 250 F, we get a flow temperature of 320 to 350 F. This gives pressures of the order of 75 to 120 psi. Where process requirements require high temperatures, this pressure may, in some cases, go up to 150 or even 200 psi.

For reasons which will be apparent later, it is desirable to fix the saturation temperature some 10 to 20 deg above that required for the flow to the system.

Avoiding Steam Formation

A typical circuit for a high pressure hot water system is shown in Fig. 1, which also gives the pressure distribution throughout the system. It will be seen that between the boiler and the pump suction, the pressure may fall below the saturation pressure, and if this occurs, part of the water will flash into steam, causing noises and shock in the piping system.

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Apart from this, there is a point at the actual suction of the pump where the water velocity is suddenly increased as it enters the impeller, where again the drop in pressure may cause cavitation.

These effects may be overcome in two ways. The pump may be placed at a level well below the boiler outlet with a connecting pipe dropping vertically to the pump suction, so increasing the static pressure. In addition, cooler return water may be injected into the flow from the boiler, as near to the boiler as possible, as shown in Fig. 2, so as to lower the temperature below the saturation point.

At the same time, in the design of the pump, special attention must be paid to the smoothness and free entry at the suction connection, so as to avoid abrupt changes in velocity.

It will be noted in Fig. 1 that the pump is shown in the flow from the boiler, and not in the return, as is perhaps more usual with low pressure systems. With the pump in the flow, the water throughout the system must be at some pressure higher than the boiler by virtue of the head created by the pump (excepting, of course, the pump suction connection referred to above). From the pump onward, the excess of pressure over the saturation pressure will naturally suppress any tendency to ebullition; whereas, had the pump been placed in the return, the boiler would be the region of highest pressure, and all other parts would be at some lower pressure—an obviously undesirable condition.

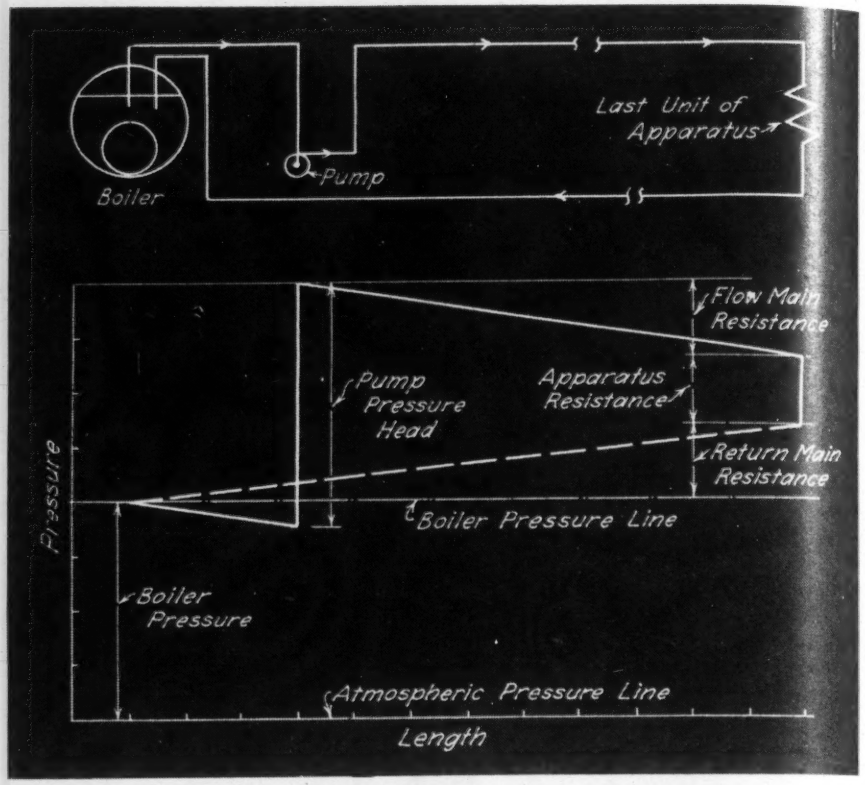

Fig. 1—Typical circuit for high pressure hot water system, showing the pressure distribution throughout it

Properties of Water at High Temperatures

The properties of water at high temperatures are not easily accessible in a convenient form, and Table 1 has been prepared to include the particular items most required over the range of temperatures in question.

From this table it will be seen that the specific heat cannot be taken uniformly at unity as at lower temperatures; also, density falls materially with increase in temperature. This has a bearing both on pipe sizes and on pump capacities.

The increase in the volume of water due to expansion may be estimated from Table 1 if the contents of the system are known. Thus, if the system, when filled with cold water at 50 F, contains 300,000 lb water, the volume will be:

300,000 multiplied by 0.01603	4809 cu ft
Volume at mean temperature of 230 F	5052 cu ft
Volume at mean temperature of 300 F	5250 cu ft
Increase from cold to hottest condition	441 cu ft
Increase from medium to hottest condition	198 cu ft

The temperatures 230 and 300 F are taken by way of example as representing the normal daily range.

Whether or not the full expansion from cold can be contained within the space of the boiler water level variation permissible depends on the type of boiler. In any event, the latter must care for the day-to-day variation between the medium working condition and the maximum working condition. The initial expansion will, in this case, have to be blown down. Generally, how-

Fig. 2—Cooler return water may be injected into the flow from the boiler to lower the temperature below the saturation point so as to avoid troubles from flashing

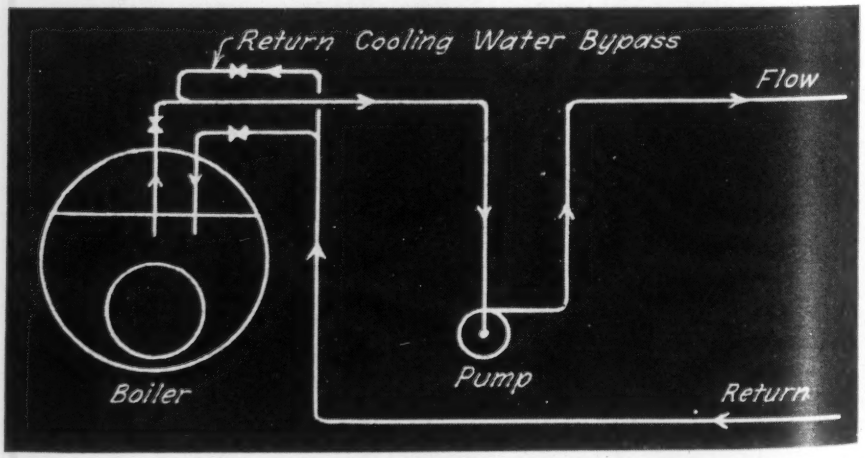

Table 1.
Properties of Water at High Temperatures

Pressure, Psig	Temperature, F	Specific Heat Btu/F	Density, Lb/Cu Ft	Specific Volume, Cu Ft/Lb	Lb per US Gal	Viscosity, Centipoises
—	50	1.001	62.41	0.01603	8.32	1.307
—	100	0.9970	62.03	0.01613	8.29	
—	150	0.999	61.20	0.01634	8.19	0.432
—	200	1.0039	60.08	0.01663	8.03	
0	212	1.006	59.77	0.01671	7.98	0.284
2.5	220	1.007	59.62	0.01677	7.96	
5	227	1.008	59.45	0.01682	7.94	
6	230	1.009	59.37	0.01684	7.92	
10	239	1.012	59.10	0.01692	7.89	
12.6	245	1.014	58.91	0.01697	7.86	
15	250	1.015	58.79	0.01700	7.84	0.195
17.8	255	1.017	58.69	0.01704	7.83	
20	259	1.018	58.56	0.01708	7.82	
23.8	265	1.019	58.42	0.01712	7.80	
25	267	1.020	58.35	0.01714	7.79	0.163
27.1	270	1.021	58.22	0.01717	7.77	
30	274	1.022	58.11	0.01721	7.76	
34.5	280	1.023	57.94	0.01726	7.73	
35	281	1.023	57.91	0.01726	7.73	
38.5	285	1.024	57.77	0.01731	7.72	0.145
40	287	1.025	57.69	0.01733	7.70	
42.8	290	1.026	57.59	0.01736	7.69	
45	292	1.026	57.54	0.01738	7.68	
47.4	295	1.027	57.44	0.01741	7.67	
50	298	1.028	57.32	0.01744	7.64	
52.3	300	1.029	57.2	0.0175	7.66	0.13
55	303	1.030	57.1	0.0175	7.66	
60	308	1.031	56.9	0.0176	7.57	
65	312	1.032	56.8	0.0176	7.57	
70	316	1.033	56.7	0.0176	7.57	
75	320	1.035	56.5	0.0177	7.48	0.115
80	324	1.036	56.4	0.0177	7.48	
90	331	1.038	56.3	0.0178	7.48	
100	338	1.041	56.1	0.0178	7.48	
110	344	1.043	55.8	0.0179	7.41	
120	350	1.045	55.6	0.0180	7.41	0.08
140	361	1.050	55.2	0.0181	7.32	
160	371	1.054	54.9	0.0182	7.32	
180	380	1.060	54.4	0.0184	7.24	
200	388	1.066	54.1	0.0185	7.24	

Table 3—Heat Emission Coefficients for Bare Piping in Btu per Sq Ft per Hr per 1 F Difference Water to Air

Temperature Difference, F	Nominal Pipe Bore, In.						
	½	1	2	4	6	8	12
100	2.54	2.38	2.22	2.06	1.98	1.93	1.86
125	2.69	2.53	2.36	2.20	2.15	2.05	1.97
150	2.86	2.70	2.51	2.33	2.247	2.18	2.10
175	3.02	2.83	2.65	2.47	2.37	2.31	2.23
200	3.15	2.97	2.78	2.63	2.50	2.435	2.35
225	3.34	3.12	2.93	2.73	2.63	2.57	2.48
250	3.47	3.28	3.07	2.87	2.77	2.70	2.60
275	3.62	3.42	3.21	3.01	2.91	2.83	2.74
300	3.78	3.59	3.36	3.16	3.05	2.99	2.89
350	4.12	3.90	3.67	3.45	3.39	3.27	3.17
400	4.45	4.24	4.00	3.78	3.66	3.59	3.49

Table 2—Percentage Corrections for Temperature When Using Pipe Sizing Data Based on about 150 F

Corrections given are positive except where stated negative, (—). Correction applies to pressure loss for a given rate of flow.

	Velocity at 150 F, Ft per Sec	Nominal Pipe Bore, In.						
		½	1	2	4	6	8	12
At 250 F	0.5	-26	-20	-13	-8	—	—	—
	1	-18	-12	-6	—	—	—	—
	2	-9	-6	—	—	—	3	3
	4	-4	—	—	3	3	4	4
	8	—	—	—	3	4	4	4
	12	—	—	3	4	4	4	4
At 350 F	0.5	-33	-24	-13	-3	—	—	7
	1	-21	-13	-4	—	5	6	7
	2	-10	-8	—	6	8	8	9
	4	—	—	6	8	9	10	10
	8	4	6	8	9	10	10	10
	12	6	7	9	10	10	10	10

Table 4—Transmission Coefficients for Radiant Heating Panels Using HPHW

Difference in Temperature Mean Water to Air	Face Downwards	Sloping, Degrees to Horizontal			Vertical
		10	20	30	
Actual Heat Emission, Btu per Sq Ft per Hr					
220	400	463	472	480	491
200	360	417	424	432	443
180	320	369	376	384	394
160	280	323	329	336	345
140	240	277	282	288	295
120	200	231	235	240	245
100	160	185	188	192	197
Heat Emission Including Comfort Effect					
Multistory Building — Room Above and Below Heated — Two Walls Exposed	Add 50% Up to Heights of 12'-0". Deduct 1% From Total For Each Foot of Additional Height	Add 30%	Add 15%	Add 3%	Deduct 3% For Height of 12'-0"
Multistory Building — Three or Four Walls Exposed, or Two Walls Exposed and Room Over Unheated or Top Floor With Two Walls Exposed	Add 38% Up to Heights of 12'-0". Deduct for Greater Heights As Above	Add 20%	Add 10%	Deduct 5%	Not Recommended
Single or Multistory Building — Three or Four Walls Exposed	Add 20% Up to Heights of 12'-0" Deduct for Greater Heights As Above	Add 10%	Nil	Not Recommended	Not Recommended

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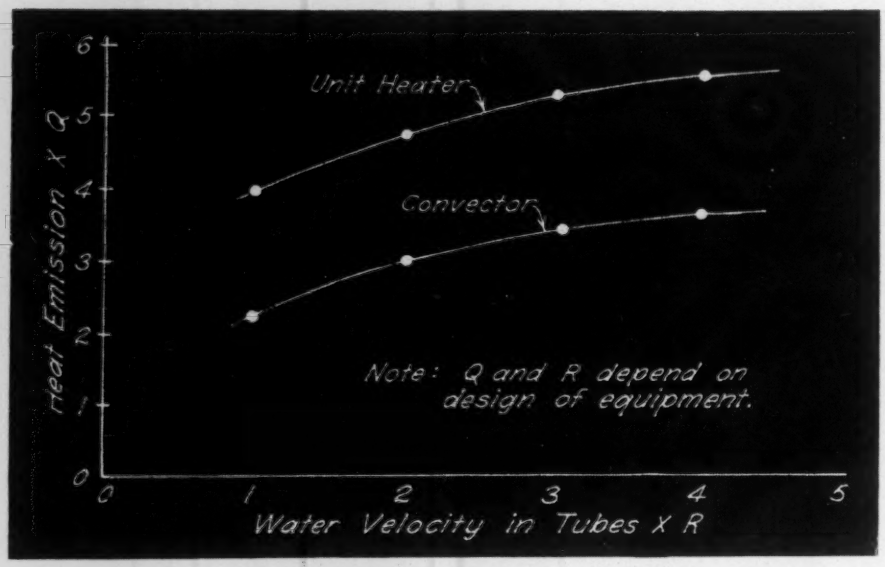


Fig. 3—Relationship between velocity of water and heat output of a unit heater and a convector

ever, where the boiler has a separate drum, it is possible to arrange for the full volume of the expansion from the cold condition to be taken care of within the capacity of the drum.

Pipe Sizing for HPHW

The tables and charts normally used for sizing hot water systems are for temperatures of the order of 140 to 180 F. With water temperatures of the order of 250 to 350 F, it appears, at first sight, that such data may no longer be applicable.

The normal rational formula for loss of heat per unit length is of the form $p = (f V^2)/(2gm)$, and for circular pipes this becomes $p = 174.2f\ (w^2v)/(d^5)$; where p = pressure drop per foot run of pipe in lb per sq in., f = a friction factor, V = velocity in ft per sec, g = acceleration due to gravity, m = mean hydraulic depth, w = weight of water flowing in lb per sec, v = volume in cu ft of 1 lb of water, d = internal pipe diameter in inches, and μ = viscosity in centipoises.

Consider first the effect of temperature change on the friction factor f. According to McAdams and Sherwood, in a paper in Mechanical Engineering, vol. 48, 1926, p. 1025, $f = 0.0054 + 0.375\ (\mu\ v)/(Vd)$.

Now (v/V) = a constant, for a given pipe size, with any range of temperature since, as specific volume v increases, so proportionately V increases also.

In this expression then, changes in viscosity μ alone affect f, for a constant diameter d. It will be seen from Table 1 that over the range 150 to 300 F, μ varies between 0.43 and 0.13, and v between 0.01634 and 0.0175.

If V is taken as, say, 3 ft per sec and pipe diameter d at, say, 2 in., at 150 F, $(v/V) = 0.01634/3 = 0.00544$ and is the same at 300 F.

Then, at 150 F, $f = 0.0054 + (0.375 \times 0.43 \times 0.00544)/2 = 0.0058$.

And at 300 F, $f = 0.0054 + (0.375 \times 0.13 \times 0.00544)/2 = 0.0055$.

If the pipe is 12 in. bore instead of 2 in. and V is again 3 ft per sec, at 150 F, $f = 0.0054 + (0.375 \times 0.43 \times 0.00544)/12 = 0.0055$, and at 300 F, $f = 0.0054 + (0.375 \times 0.13 \times 0.0054)/12 = 0.0054$.

Thus, as pipe diameter increases, the effect of changes in viscosity becomes less and less. This is, of course, also clear from the normal curves of Reynolds numbers plotted against friction factor, in which, after turbulent flow has been established, the curve tends to flatten out for the higher Reynolds numbers, particularly with rough pipes.

Considering now the effect of temperature change on the rest of our second formula, we may regard the weight of w as constant. The term v (specific volume) varies, from 0.01634 at 150 F to 0.0175 at 300 F.

If we take the combined effect of temperature change on f and v, we have (for $d = 2$ in., $w = 3$ lb per sec, and p' = pressure loss in lb per sq in. per 100 ft):

At 150 F, $p' = 174.2 \times 0.0058 \times [(3^2 \times 0.01634)/2^5] \times 100 = 0.465$ lb per sq in.

At 300 F, $p' = 174.2 \times 0.0055 \times [(3^2 \times 0.0175)/2^5] \times 100 = 0.472$ lb per sq in.

For $d = 12$ in. and $w = 300$ lb per sec, at 150 F, $p' = 174.2 \times 0.0055 \times [(300^2 \times 0.01634)/12^5] \times 100 = 0.566$ lb per sq in.

And at 300 F, $p' = 174.2 \times 0.0054 \times [(300^2 \times 0.0175)/12^5] \times 100 = 0.595$ lb per sq in.

The differences in these examples are, for 2 in. pipe, 1.5 percent and, for 12 in. pipe, 5.1 percent. Table 2

Top: Fig. 4—Heating element designed with parallel rows of tubes for steam. Bottom: Fig. 5—A series flow as shown here is more desirable than the parallel flow of Fig. 4 for use with HPHW

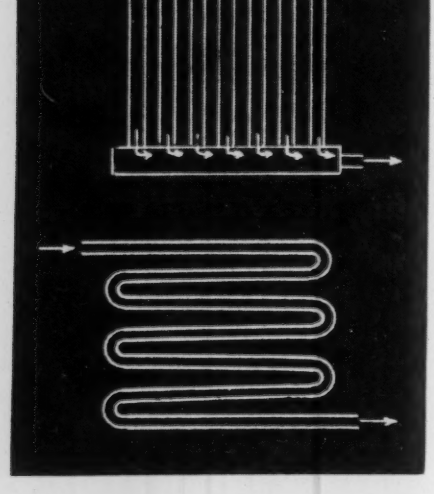
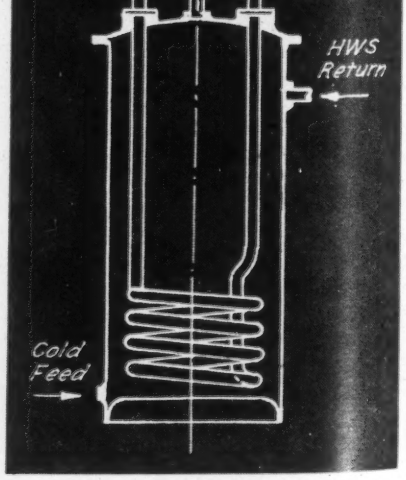


Fig. 6—Water heater with continuous spiral coil for use with HPHW



gives values for these variations for temperatures of 250 F and 350 F according to pipe diameter and velocity. No difference is given less than 2 percent.

When making use of the correction percentages given in Table 2, it is well to bear in mind one or two facts in connection with pipe sizing generally, which make extreme accuracy unobtainable in any event.

All pressure loss formulas involve a friction factor, which has been variously determined by numbers of investigators, mostly using small pipes. Extrapolated values have generally been calculated for bigger sizes. In comparing, for instance, the work of Rietschel with the ASHVE's Guide and the British IHVE Guide, it will be found that there are commonly differences of the order of 10 percent in flow rate for a given pressure drop for the smaller sizes, and as great as 30 percent for larger sizes.

Again, in the matter of commercial pipe sizes, it can be shown that permissible manufacturing tolerances in diameter alone may account for a variation in pressure drop of as much as 15 percent in certain sizes.

Thus it is clear that there is no point in practice in entering into a highly involved mathematical exercise for arriving at pressure drops with these higher temperatures, when we are faced with much greater inaccuracies due to our present imperfect state of knowledge or manufacturing facilities.

In general it may be said that if a designer is in the habit of using a certain set of data for pipe sizing hot water systems at normal temperatures, he can quite safely continue to use it for high pressure hot water, provided he checks over the pressure drop carefully before selecting the pump head, applying corrections as given in Table 2. Alternatively, if he has access to tables or charts prepared expressly for the conditions in question he will obviously be well advised to use them.

Heat Emitting Surfaces

The heat emission from bare piping heated by HPHW may be calculated in a similar manner to surfaces heated by steam, and Table 3 gives coefficients that the author has found applicable at the various temperatures commonly applied.

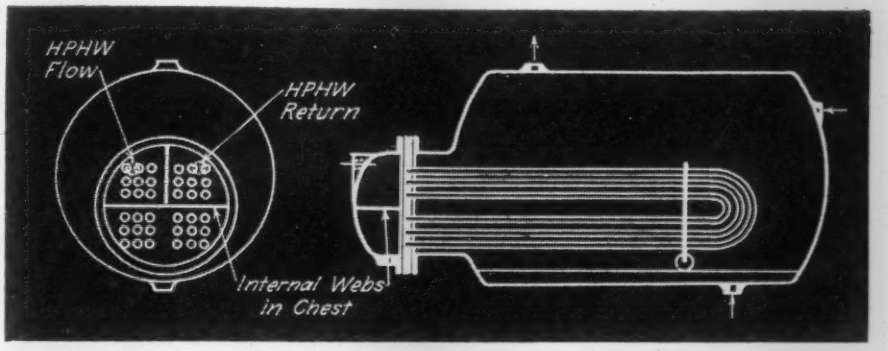


Fig. 7—Series parallel form of water heater for HPHW use

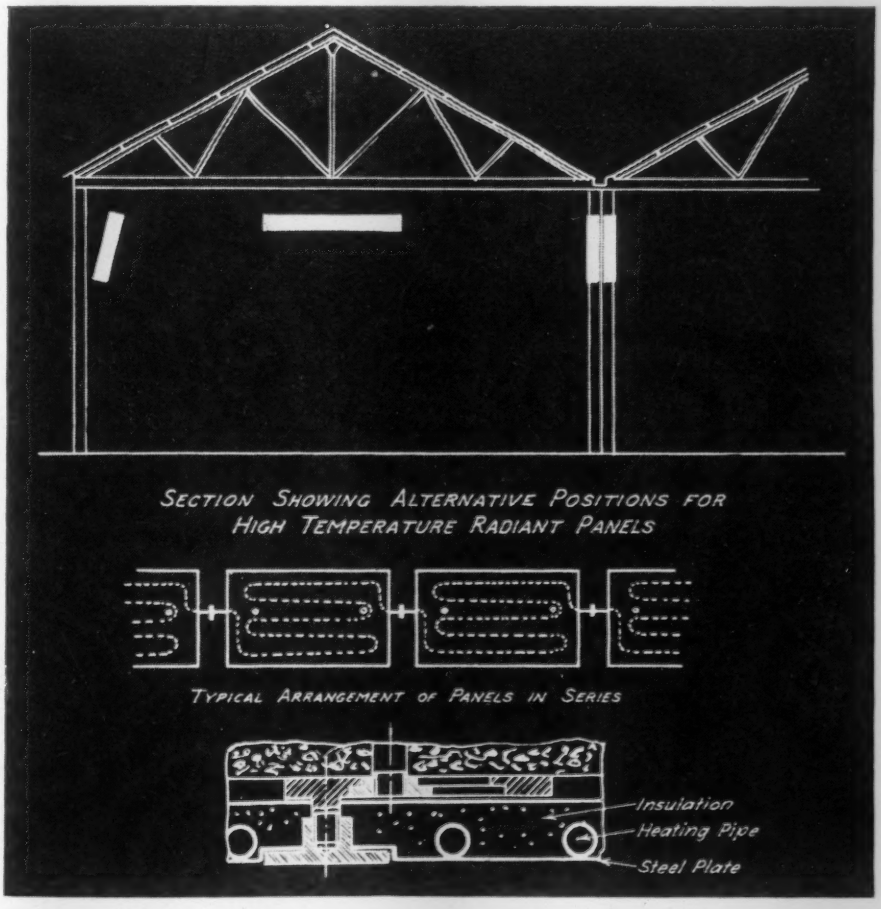
Where piping is insulated, the loss may be estimated by taking the bare pipe loss multiplied by the appropriate loss factor through the insulation adopted.

Most of the heat emitting surfaces in a high pressure system will be of the finned tube type with forced air circulation over them, such as unit heaters, convectors, etc., and as the types and varieties vary so widely, it is not possible to give here data which would be of service.

It is, however, possible to indicate the importance of water velocity in the design of such apparatus. Fig. 3 is a curve based on manufacturers' tests of a unit heater and a convector, showing the relationship between velocity of water and heat output, at a constant mean temperature in each case.

It will be seen that the heat output may be increased appreciably by raising the flow rate. Thus, if for steam, a heating element is designed with several parallel rows of tubes, as in Fig. 4, it would be desirable for use with hot water to arrange for a series system, as in Fig. 5. The pressure drop will naturally be raised, but this may be allowed for in calculating the pump head. After all, some sizable resistance in the heat emitting elements is a help in rendering the system more nearly

Fig. 8—A patented form of radiant heating panels used with HPHW


Section Showing Alternative Positions for High Temperature Radiant Panels
Typical Arrangement of Panels in Series
Insulation
Heating Pipe
Steel Plate

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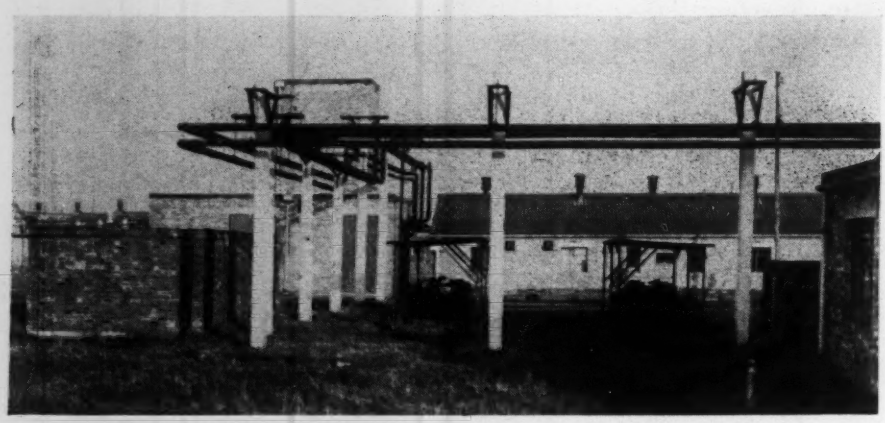

Overhead HPHW lines entering a water heating substation

self balancing without undue regulation of valves.

It should be mentioned that heating surfaces suitable for steam are not always suitable for water circulation, partly because in the case of steam the trapping of air in pockets near the top of heaters, etc., does not arise and partly because in the case of hot water, the design of the headers must be such as to provide a uniform high velocity through the tubes.

What has been said of finned surface elements for air heating applies similarly to water-to-water transmission surfaces, as in water heaters and other heat transfer coils. Water heaters for HPHW are commonly arranged with a coil in continuous spiral form, as in Fig. 6, or in a series parallel form using U tubes, as in Fig. 7. The same reasons for maintaining an optimum velocity in the tubes naturally apply with all such apparatus.

Radiant heating with HPHW may take the form of metal plates with pipes welded to the back in continuous sinuous coils. One patented form of this type of surface is shown in Fig. 8.

These panels may be fixed horizontally overhead in workshops, factories and the like, provided the mounting height is adjusted so as to avoid an unpleasant intensity on the head. About 14 to 15 ft is suitable, with water at about 280 F mean, the top surface being insulated.

Alternatively, the panels are sometimes fixed at an angle along the sides of the shop and sometimes vertically back to back down a center row of columns.

Transmission coefficients may be taken from Table 4, according to the author's experience.

Stabilization of Stainless Steel

Austenitic stainless steels of the type usually referred to as 18-8 (18 percent chromium—8 percent nickel) are sometimes susceptible to intergranular embrittlement or corrosion. This type of corrosion is particularly pronounced in some 18-8 steels which have been subjected to elevated temperatures—in the range 700 to 1400 F—and are either simultaneously or subsequently subjected to corrosive conditions. Experience has shown that the susceptibility to embrittlement may be decreased or eliminated—that is, the steels may be stabilized against intergranular embrittlement—by the addition of titanium or columbium, usually in conjunction with a stabilizing heat treatment.

A diversity of opinion has existed, however, as to the relative amounts of titanium or columbium necessary for effective stabilization of these steels, the injurious effect of carbon content, and the necessity for stabilizing heat treatments. Accordingly the National Bureau of Standards was requested by the Navy Department bureau of aeronautics to undertake a study of factors affecting the stabilization of the 18-8 type of steels.

Using both experimental and commercial steels made to a base analysis of 18 percent Cr and 10 percent Ni, the effect of variations in carbon content, in the ratios of columbium to carbon and of titanium to carbon, and of different heat treatments, was studied by Samuel J. Rosenberg and John H. Darr of the NBS thermal metallurgy laboratory*.

Considering the steels which contained no stabilizing elements, the tests showed that all were vulnerable to intergranular attack. Decrease in carbon content, however, decreased the degree of vulnerability.

In the columbium- and titanium-bearing steels, carbon content within the range of 0.06 to 0.13 percent had no influence upon the resistance to intergranular attack except as it influenced the Cb/C or Ti/C ratios. Steels varying in carbon content but having similar ratios of Cb/C or of Ti/C had approximately the same degree of susceptibility to intergranular attack regardless of the total carbon content. The steels showed greater resistance to attack when annealed at 1800 F than when annealed at 1975 F.

Stabilizing heat treatments at 1600 F had a negligible effect upon the resistance to intergranular embrittlement of the columbium-treated steels so that these steels carrying a sufficiently high ratio of Cb/C may be used without giving them a stabilizing heat treatment. However, the performance of the titanium-treated steels carrying the higher ratios of Ti/C was markedly improved by such treatments. When properly treated, substantially complete immunity to intergranular attack may be obtained with a minimum ratio of Cb/C = 10 and Ti/C = 5. For more "foolproof" immunity, these ratios should be 12 and 8, respectively.

* For a complete technical account of this work, see Stabilization of Austenitic Stainless Steel, by Samuel J. Rosenberg and John H. Darr, Journal of Research, National Bureau of Standards, RP1878, April 1948.

QUESTION OF THE MONTH

YOU ARE invited to submit a heating, piping, or air conditioning question for publication here. You are also invited to submit answers to previously-published questions from others. Those published are paid for at regular rates. Address the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.

BUSINESS ADVANTAGE OF AIR CONDITIONING

IN THE APRIL issue, "J.O." asked what percentage increases in business can be expected in a store, a restaurant, and a hotel dining room due to the installation of a good summer air conditioning system.

It might not be impossible to answer his question numerically with a fair approximation, though this would involve statistical investigations on a large scale. It is not the scope of this contribution to offer anything like definite values, but only to give a hint how—lacking sufficient statistical data—the question could be approached.

If a store, a restaurant, or a hotel dining room is provided with a suitable air conditioning plant, the number of customers will, no doubt, increase, and the turnover and the profit will rise. If at the time of installation all other (or the majority of) similar businesses in the town already had air conditioning systems, the owner of the specific company will find that he recovers the loss suffered before due to the inferiority of his facilities. The installation then represents a necessary measure to keep pace with the acknowledged progress of the trade, and the economy is given by the return to normal success.

We must keep in mind that air conditioning means cooling in summer and heating in winter. The time is not far when both will be considered as equally desirable. And who would risk to open a store, a restaurant, or a hotel dining room without a heating plant?

Things seem to be different, if no air conditioning systems are in operation in similar businesses of the town. The installation will, then, secure superiority and increase in profit. Still, such success will be only temporary, and the competition will, willingly or unwillingly, soon follow the example and regain balance.

My answer in general is, therefore, that an air conditioning plant for the mentioned applications is an integral part of the necessary equipment, provided, of course, that local weather conditions are such as to present discomfort in summer unless artificial means are applied.

These remarks consider only the psychological effect on customers, who prefer to visit places with more comfortable conditions. But the economical success of air conditioning does not end with the number of attracted customers. If this were the case, the totality of similar businesses would only preserve their standing as before, and the owners could be inclined to come to an arrangement not to invest capital for a nonpaying improvement. Actually the whole trade profits by air conditioning, particularly in two aspects:

In air conditioned restaurants and hotel dining rooms, patrons will take normal meals, whereas in the close summer atmosphere of a nonconditioned space, light refreshments will be preferred. If a hotel has no air conditioning plant and a nearby restaurant has, the hotel guests will prefer to take their meals in the restaurant.

For a store, relations are more complex and depend on the kind of goods on sale. An illustrative example is given by a fur department. No lady will undergo the ordeal of trying and buying a fur coat in summer in an unconditioned hot store, but she might do so in the balmy atmosphere of a well conditioned store. The result is a welcome balancing of fluctuations in business, an artificial extension of the seasons, and more profit for the storekeeper.

The second aspect concerns the staff. Air conditioning renders employees and management more efficient and more able to please the customer. An exhausted mannequin does not make an inviting show, and a perspiring waiter does

How Keep Top Floor Comfortable?

You are invited to comment on the following question from "C.H.S." Suitable comment on this matter will be published, and paid for at regular rates (with additional payment for sketches). Address The Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2.

"May I submit the following as a Question of the Month:

"I have a case where an upper story room measuring 80 by 80 ft is to be made comfortable. The ceiling—12 ft high—is formed by rafters and boards, painted white on the underside and covered on top by two layers of black roofing. Windows are on the south and west sides, amounting to 38 percent of that wall space, and they have venetian blinds.

"The room is at present occupied to about one-half of the floor space by 70 drafting tables, by two glass enclosures 12 by 12 ft, open on top and occupied by two seated office workers each. About 120 sq ft is taken up by filing cabinets. The rest is empty. The atmosphere is strictly informal, no coats are worn.

"The ceiling gets hot when the sun shines. While I know how many Btu's to figure for sun load on the 6400 sq ft roof area, I do not know how hot the ceiling will be and how to compensate for its radiation into the room. The cost must be kept as low as possible, and the equipment should not become part of the building because the space is leased. Insulating the roof is therefore not given consideration.

"I wonder if evaporative coolers will be at all sufficient. If they are, how many cfm should they be able to handle?"—C.H.S.

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not encourage guests to extend their stay. The sensitive may be disgusted and never come back. And, we must not forget the fact that absenteeism through illness of the staff is greatly reduced in a business which possesses a perfect air conditioning plant.—H. M.

CORROSION OF CLOTH TYPE FILTERS

THE FOLLOWING remarks are in answer to a question from a reader about the corrosion of filter fabric in cloth type filters when handling dusts with an acid reaction.

First of all, a cloth filter is fundamentally and basically adaptable to the collection of fine, dry dusts entrained in air with a normal moisture content. Moisture and a cloth filter do not go together whether the moisture is contained in the dust, in the air, or in both. It is our experience that most of the corrosion in cloth filters handling materials having an acid reaction is caused by moisture condensation which accelerates the corrosive condition. If the dust and the conveying air can be kept dry and the collector temperature maintained well above the dew point at all times, the corrosive action will be reduced.

It is very likely that the corrosive condition in the crushing plant filter mentioned by your reader is accelerated during colder weather and particularly when the air humidity is high. Exhausting air from a heated plant to a filter located outdoors can result in considerable moisture condensation during colder weather, particularly when the humidity of the air is high. If moisture is liberated from the ore being processed in the plant, the condition may be further aggravated.

Woolen filter fabric is slightly more acid resistant than cotton fabric. Also, some of the newer fabrics woven from manufactured fibers are more resistant to acids. However, unless extreme care is exercised in controlling the temperature and moisture, particularly when collecting dusts with an acid reaction, not only will corrosion occur but the cloth filter cannot perform its function properly and effectively.—W. O. VEDDER, dust control dept., Pangborn Corp., Hagerstown, Md.

LONG CONCRETE RAMP KEPT FREE OF SNOW

To facilitate movement of materials and finished products between plant buildings by industrial trucks, the plant engineering staff at the Bigelow-Sanford Carpet Co., Inc., factory in Amsterdam, N. Y., designed an outside 85 ft long concrete ramp leading from the basement of one building to the street.

Topography of the area made it necessary to build the ramp on a 10 percent grade and install a 30 deg turn. The problem, however, was not so much how to construct the ramp as it was to decide what to do about its use during winter.

In that part of New York state, snowfalls are frequent and heavy. Snow or ice on the ramp would obviate its use by industrial trucks and delay production or shipping operations. Keeping the ramp free of ice and snow would constitute a major maintenance item.

The problem was solved by designing and installing a network of pipe coils in the 8 in. crushed rock fill underlying the 6 in. concrete slab. Hot water, heated by a converter on the plant's steam line, is forced through the coils by a pump to keep the pavement warm and thus melt snow as quickly as it falls and prevent ice from forming. Last winter, snow depth ranged from 18 to 30 in. but the system worked to perfection, plant officials say. The pavement remained clean and dry without the use of snow shovels or chemicals. Fuel costs were estimated to be low although no definite check was made.

At the top and bottom of the 9 ft wide ramp, catch-basin type drains are installed to carry away residual water and normal storm drainage. The drain pipes are 8 in. wrought iron. Heating pipes also are run through the catch basins to minimize possibility of the drains freezing. Antifreeze is added to the water used in this closed system to prevent freeze-up during non-operating periods.

The wrought iron pipes are welded together in the form of a grid. They were cold bent where necessary to follow the slope and curve of the ramp. Header pipes and supply and return mains are 2 in. and the grid pipes, spaced 13½ in. apart between the headers, are 1¼ in. pipe. With this pipe spacing, and an operating water temperature of 130 F, the system is designed to melt snow at a rate of 2 in. per hr. If the system is turned on when snow starts to fall, the temperature of the pavement is raised sufficiently to melt the flakes upon contact.

A vent line, with a manually operated air valve, is installed in the high point of the piping layout.

Grid header pipe at foot of ramp. Grid pipes run through catch basin to prevent drainage water from freezing. Residual water from melted snow and normal rain drainage is carried off through 8 in. drain pipe

"OPEN FOR DISCUSSION"

DO YOU HAVE something to say about any of the articles published this month? An additional comment, or perhaps a question on some point? We follow here the custom of engineering society meetings in allowing a period for discussion following presentation of a paper. You are invited to contribute your views. Address the Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.

PRACTICAL POINTERS ON RADIANT HEATING

WE HAVE READ with interest R. G. Vanderweil's radiant heating article in the March HPAC and feel called upon to make some observations on his conclusions. In view of the fact that we are involved in radiant heating from both design and operational standpoints, our comments may be of interest to HPAC's readers.

Mr. Vanderweil first discusses the ceiling vs. floor panel situation and suggests that floor panels be used for rooms having ceiling heights exceeding 15 ft. We have observed a ceiling panel installation made in an industrial plant, with a ceiling height of 22 ft. The temperature of the unheated floor slab was 12 F above that of a floor in an adjacent bay (heated by radiation). Since the performance of this particular installation was completely satisfactory in all respects, Mr. Vanderweil's restriction seems too severe.

It would be better, we think, to suggest that floor panels be used where floors are concrete, *if* floor coverings will not be felt pads and wall to wall rugs. Needless to say, if floors are the conventional wood type, superior results will be obtained from ceiling panels.

Concerning lag or heat storage of panels, our experience to date has disclosed the following:

Ceiling panel installations are no more difficult to control than conventional water systems.

Conventional controls (air temperature thermostat and boiler aquastat) produce satisfactory conditions with floor panel installations.

There will be no appreciable lag with sudden drops in outside temperature, and while a recording thermometer will show an overshoot with a sudden rise in outside temperature, the overshoot will not be sufficiently great to cause discomfort to the occupants. At the time of writing, there are approximately 800 Canadian radiant heating installations, and it is interesting to note that we have not yet had occasion to service one installation equipped with conventional controls.

Mr. Vanderweil's observations on basementless structures do not seem to bear out the results of our experience. Since radiant heating was introduced to Canada, we have been particular about such installations in only one respect; the breaking of the slab by an insulating strip around the perimeter.

Results to date indicate that the basementless structure requires less fuel than the conventional one, and in every case where fuel consumption has been estimated, it has been found that the actual consumption was slightly less than the theoretical, even though losses down from the slab were not included in the calculation. ASHVE methods were used, and while these may be generous—thus allowing for a "hidden" floor loss—the floor loss could not have been great.

Heat Loss to the Ground

In designing, we have been using an arbitrary figure of 15 percent of all other losses to cover the heat escaping to the ground, and this has proved adequate for outside temperatures down to —15 F.

Where an installation goes into operation for the first time in freezing weather this allowance is inadequate, but after some hours of operation, the slab comes up to temperature and no lack of capacity will be noticed from that point on.

Because of the foregoing, we are not yet convinced that insulation under slabs is necessary, unless unusual conditions are encountered—such as a high water table. We do agree, however, that slab edges should be insulated and that such insulation should be carried well down the inside of footings (2 ft seems a reasonable minimum).

As regards venting, this should not be a point of concern for basementless structures, where boilers are above or on the same level as floor coils. One vent at the boiler is sufficient. For two or three story installations, we agree with Mr. Vanderweil, although our experience indicates that hand operated vents are preferable.

In conclusion, may we add that the foregoing is in no way intended to criticise Mr. Vanderweil's conclusions, although his article prompted it. We have intended for some time to send these observations to you, since some of them are at variance with those expressed in HPAC from time to time.—W. A. CONNOR, service engineer, radiant heating div., Page-Hersey Tubes, Ltd., Toronto.

CODE RESTRICTIONS ON WELDING

WELDING CONSTRUCTION is being handicapped by code writers putting restrictions of no value, but of great expense, on welded structures made under their codes. The following illustrations will show some of the distortions used to limit welding and to increase its cost.

It is impossible in mild steel to make a welded joint of equal sections which is not stronger and will not break the plate in tension. However, the code writers, in spite of these facts, make the following restrictions on welded structures.

If a welded joint is made of equal strength to a riveted joint, the riveted joint is accepted, the welded joint rejected.

If a welded joint is made of equal ductility to a riveted joint, the riveted joint is accepted, the welded joint rejected.

If a welded joint is made with the same undercut from calking, as is normally done in a riveted joint, the riveted joint is accepted, the welded joint rejected.

If a welded joint is made with the same porosity as a riveted joint, the riveted joint is accepted, the welded joint rejected.

If magnaflux or x ray is used on a welded joint and it shows the same voids as it would on a riveted joint, the welded joint would be rejected, the riveted joint accepted.

If a welded joint is made with the same surface appearance as a riveted joint, the welded joint would

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be rejected, the riveted joint accepted.

If a welded joint is made with the same tightness as a riveted joint, the riveted joint would be accepted, the welded joint rejected.

If a welded joint is made with the same fatigue value as a riveted joint, the riveted joint would be accepted, the welded joint rejected.

If a welded joint is made with the same tendency toward caustic embrittlement as a riveted joint, the welded joint would be rejected, the riveted joint accepted.

There has never been a weld failure in any insured pressure vessel. In spite of this, progressively greater handicaps are placed on welding whose sole purpose is to make welded structures cost more. Already there have been billions of dollars unnecessarily wasted in this way.

Perhaps what we need is a new code for welding code writers rather than for welded constructions. Perhaps we can then deal with reality in this matter.—JAMES F. LINCOLN, president, The Lincoln Electric Co., Cleveland.

LECTURE AVAILABLE ON COMPRESSED AIR

The committee on engineering education of the Compressed Air and Gas Institute, 1404 Terminal Tower, Cleveland, has announced a 30 min illustrated lecture on compressed air power and its modern industrial applications.

Produced primarily as a teaching aid for engineering schools and colleges, and reviewed by a group of engineering professors, Compressed Air in Action contains 61 illustrations in either 35 mm continuous strip film or 2 by 2 in. film slides. The lecture is available at cost to industrial organizations, technical societies, high schools, vocational schools, foremen's clubs, service clubs, and any other interested groups.

Purpose of the lecture—which is accompanied by a 20 page informational brochure containing reproductions of the slides—is to illustrate and describe modern basic types of air compressing equipment and air operated tools and rock drills, and to portray compressed air power in operation in many typical present-day services.

Equipment illustrated and described includes reciprocating, portable, rotary and centrifugal compressors; percussion tools; and rotary tools. Application photos show modern compressed air applications in construction, manufacturing, mining, agriculture, food processing, transportation, and maintenance.

7755 BETTER WAYS TO DO THE JOB

Employees of the Westinghouse Electric Corp. during 1947 figured out 7755 new and better ways to do their work, the company has reported. T. O. Armstrong, director of plant labor relations, said a total of 20,743 ideas were sent in through the suggestion system. Of these 7755 were adopted, bringing a total of more than $105,000 in awards to some 5000 employes.

Largest award of the year—and highest since the suggestion system was begun in 1910—was $5000 paid to Alexander McLennan, repair shop foreman. He designed and built a special transformer coil unit that fit into all old transformer tanks of the type used in residential areas. His suggestion enabled repair and reconditioning of transformers in one-quarter the time once required.

Mr. McLennan worked for nearly two years in his spare time to perfect his coil unit. Most suggestions, however, did not require this much work, Mr. Armstrong said, but stemmed directly from the employe's familiarity with his own job.

INDUSTRIAL AIR CONDITIONING—

[*Concluded from page 96*]

Check depreciation rate of each, allowing for depreciation of eliminators and spray nozzles with washer.

Check maintenance. Coil may get slimy and require cleaning. With washer, nozzles will clog and eliminators will require replacement.

Check operating costs. Here the coil may have the advantage.

Check space requirement for each type of equipment.

Check water required for washer. Temperature rise in washer is limited. Quantity also depends on number of spray banks.

Check pump required with washer.

Check air filters required. Washer will not eliminate smoke and soot.

Check on mean temperature difference and its effect with each type of equipment. Here, the direct expansion coil has an advantage.

Check on allowable distance from refrigeration machine with coils (limited) and from water chiller with washer (not limited).

Check on control of humidity, which is limited with coils. Coils usually deliver unsaturated air, but do more dehumidification than washer. If washer delivers saturated air, humidity control can be excellent, especially where narrow limits of relative humidity are required.

Direct Expansion Coils and Chilled Water Coils

In making a choice between mechanical cooling with direct expansion coils and with coils using chilled water, consideration is given the following items.

Check initial cost. For the same amount of surface, a direct expansion coil costs more but it may need less surface for a given load. With chilled water coils, cost of the pump is the major difference. The cleanable type of water coil will cost more than the standard type used for direct expansion.

Check installation costs. Here it is a water pump and piping against refrigerant piping and fittings.

Check depreciation rate, maintenance, and space requirement, with little to choose between either type, except if cooling water contains minerals which will scale the tubes or dirt or sand which will necessitate frequent cleaning.

Check water required with chilled water coil, the amount depending on the temperature rise.

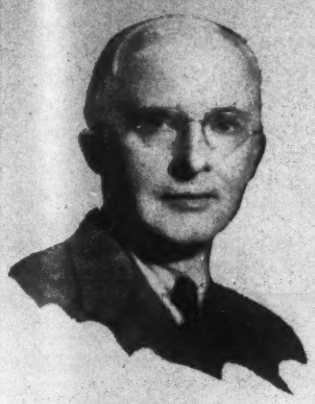
Check operating costs. Here the direct expansion coil may have the advantage.

Check effect of one heat transfer required with a direct expansion coil and two transfers with chilled water coil.

Check mean temperature difference possible with each type.

Check on distance from refrigeration machine (limited with direct expansion coil) and distance from water chiller (not limited with chilled water coil).

Check on control of humidity with each type, which is limited. Coils deliver unsaturated air. Direct expansion coil usually does more dehumidification work.



SAM LEWIS' PAGE

Informal comment on heating, piping, and air conditioning matters is given in this regular feature by Samuel R. Lewis, consulting mechanical engineer, member HPAC's board of consulting and contributing editors

Don't Neglect Zoning With Solar Heat Jobs

While a conventional building having four-side exposure may be livable as far as winter temperature is concerned when controlled as one zone, buildings fitted with glass walls on the sunny side to utilize solar heat definitely should have special zoning for temperature. An actual case in point is given here . . . . . .

THERE IS AN increasing number of buildings in which an attempt is made to conserve and to utilize the considerable heat that comes from the sun. In winter, the sun can "see" into rooms on the south sides of buildings, and if the walls are largely transparent—as ordinary glass—and if there are no clouds, surprisingly little fuel is required for such rooms during the hours of solar reception. It has not been demonstrated conclusively that such intermittent heat reception will permit any reduction in the capacity of the usual fuel burning heat sources or any reduction in the size of the local heat transmitters.

An ultramodern single story building visited in April has underfloor radiant hot water heating. There is no basement. The fuel is gas, burned in a sectional cast iron boiler designed especially for that fuel.

The piping design is decidedly favorable, with the flow main running down from the boiler to loop around the exterior walls under the concrete slab. There is an adequately proportioned serpentine coil in the concrete for every room, with a separate return pipe from each coil running separately all the way back to a header in an accessible floor pit near the boiler. Each return pipe has a square head cock capable of adjustment so as to permit equalization of the water flow.

An outdoor thermostat, originally shaded under the eaves on the north wall, controls a motorized valve in the discharge main from the electric circulating pump so as to divert the water through the boiler, or to bypass the water around the boiler in varying proportions, so that the water temperature in the flow main going to the rooms will be inversely proportional to the outdoor temperature. This thermostat had been moved so as to be exposed to the sun above the roof.

The pump was designed to operate continuously and an aquastat in the main flow pipe was designed to take over the bypass control in case the outside thermostat should call for too-cool water. The gas burner, except for the pilot flame, responded to an aquastat in the boiler water.

The longer axis of the building is east-west, and the more important rooms are on the south side, with their south walls practically all of glass.

South Rooms Warm, North Rooms Cool

At about 1:30 p.m. on a clear day the floors of the south rooms, all of which happen to have a black rubber tile surface, were brilliantly illuminated by the sun for about 4 ft

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into the rooms. The floors in the sunlight were exceedingly hot to the hand, though the shaded areas of the floors were cool to the touch. Thermometers on the shaded inner walls at breast level registered 82 deg. The water circulating pump had been stopped, by opening the motor switch, for over 2 hr. The rooms on the north side of the building were cool, especially as affecting the senses of a person entering them from the south rooms.

A thermometer in a room having north and east exposure registered about 60 deg. The outdoor temperature was 40 deg.

Patience—and abraded knuckles—had failed to accomplish satisfactorily useful throttling of the 15 return valves at the header so as to achieve equal heat distribution to all rooms at all times. The gas consumption was alleged to be excessive, but computations of this on a degree-day basis for March checked within less than 10 percent of what it should have been.

The complaint of too high temperature for the south rooms and of too low temperature for the north ones was undoubtedly valid. To protect the health and morale of the inhabitants it had been necessary to play with the switches and to cut out the automatic controls during many hours of the frequent sunny days. Constant manipulation of the sun shades on the south walls with resultant exasperating variations in light and frequent use of electric lights had awakened resentment by the inhabitants.

Special Zoning Needed

It appears that while a conventional building having four-side exposure may be livable as far as winter temperature is concerned, when controlled as one zone, a building fitted with glass walls on the sunny side definitely should have special zoning for temperature. A building with radiant floor heating exclusively, through large masses of concrete, especially requires zoning by thermostatic means to anticipate as far as may be possible the overheating and underheating in response to the variations in solar influence.

My prescription in this building was to install a thermostat on the shaded inner wall of each south room, this thermostat to control a valve in the respective return pipe from the floor coil. The original outdoor thermostat, if moved back out of the direct sunlight, might continue to control the general water delivery temperature and should restore comfort to the people in the north rooms. The new south room thermostats would stop the water circulation in these rooms if the sun was too busy, and would restore circulation locally if the sun was put out of business by clouds or by nightfall.

I was told—but unfortunately didn't believe it—that complaints like those made to me concerning this building had been resolved by up-ending a milk bottle over the outdoor thermostat exposed directly to the sunlight.

I told the fireman-custodian of this building—who also unfortunately didn't believe me—as we sat on the floor with our hands on the sun-heated area, that the water circulated through the pipes in that area would go back to the boiler and help warm the coils under the north rooms.

The switch for opening the motor circuit for the pump in plants of the type described above really ought to be hard to find—since continuous water circulation, day and night, should be maintained.

A gas burning boiler should by all means be placed in a separate room to which a copious supply of outside air can be maintained. Failure to do so may result in wet chimney walls, rusted pipe surfaces, excessive fuel consumption, and lost friendships.

LABOR TAKES STEPS TO BOOST OUTPUT

Labor organizations in the building industry recently have taken a number of important steps to increase productivity by bringing labor practices in closer accord with modern building techniques, according to a study by economists of the Construction Industry Information Committee, Melvin H. Baker, its chairman, announced last month.

"Many international unions have moved to eliminate ingrained industrial practices that have retarded technical progress, restrictions on the numbers of apprentices have been greatly relaxed, and cooperative machinery has been set up to settle jurisdictional disputes," Mr. Baker said.

"The plumbers' and pipefitters' union has emphasized its consent to the cutting and threading of pipe at the contractor's shop by its members and to their use of power tools for cutting and threading pipe of any size; the International Brotherhood of Electrical Workers has stated its approval of the use of power tools and labor saving devices; the carpenters' union imposes no restrictions on the use of power tools by its members, provided they are used under conditions safeguarding the worker; the International Brotherhood of Painters permits the use of spray guns where adequate provision is made for protecting the health of workers, and has removed objections to the installation of window sash which have been glazed in the factory.

"Labor unions in almost all crafts are cooperating with management in increasing the number of apprentices and in providing for their training, with the result that the number of apprentices in the building trades now is three times greater than it was two years ago. Fully 115,000 apprentices now are in training and efforts to expand the program are continuing.

"The president of the Bricklayers, Masons and Plasterers' International Union has urged all local unions to permit as many new apprentices as are needed to meet future building needs.

"Costly delays in construction arising from disputes between various unions as to which should handle various parts of a building job should be greatly reduced by recent developments. The AFL building and construction trades department and employer associations have devised a new plan for settling these jurisdictional disputes which promises to be generally acceptable both to management and labor.

"A basic attack on the causes of jurisdictional disputes has been made in the pipe trades where a combination of the crafts of plumber and pipefitter (steamfitter) has been undertaken by the international union and in some localities is already complete."

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Fan Selection by Use of Constants

By Arthur C. Stern* and Leon D. Horowitz**, New York, N. Y.

SUMMARY—This paper presents a simple method of applying fan constants in establishing the rating of fans in the field when fan rating tables are not available. One table of constants replaces many pounds of rating tables such as would be required to provide equivalent information.

Introduction

In March 1936, G. E. McElroy published as U.S. Bureau of Mines Report of Investigations No. 3298 a paper entitled, "Charts for Determining the Performance of Centrifugal Fans." This paper reviewed at great length all the previous attempts at the simplification of the means of fan selection and fan rating, and explored a number of new and original approaches. To anyone interested in the application and use of fans, this paper is an inspiration to build upon the groundwork so thoroughly laid. Starting with this paper as a basis the authors have investigated a number of methods of simplifying fan rating and selection. The one presented in this paper is believed to be the simplest and most useful for field application by the industrial hygiene engineer. The method described makes use of fan constants for the selection and rating of fans without reference to the conventional manufacturers' fan rating tables. The method presented is not intended to replace the use of fan rating tables, but rather to supplement their use, particularly where it is inconvenient to carry a full set of such tables into the field.

The Division of Industrial Hygiene and Safety Standards has collected data for each of the principal makes and types of fans to enable the use of the method presented. The Division has available for distribution to those interested tabulations of the fan constants pertaining to the principal fans encountered in the field. This information has been prepared with the cooperation of all of the principal fan manufacturers.

Development of Method

The following letter symbols are used in the equations which follow:

- A = area of fan inlet, square feet
- A_o = area of fan outlet, square feet
- C = a constant
- D = fan inlet diameter, inches
- d = fan wheel diameter, inches
- E = fan static efficiency (a decimal less than 1)
- H = static pressure, inches of water
- K = a constant
- N = fan speed, revolutions per minute
- P = fan horsepower input, actual
- Q = air quantity, cubic feet per minute
- V = velocity at fan inlet, feet per minute
- V_o = velocity at fan outlet, feet per minute
- W = fan casing width, inches

The principal fan laws may be stated as follows:

$$\begin{aligned} Q &= C_1 N; \quad H = C_2 N^2; \\ P &= C_3 N^3; \quad E = K; \\ D &= K \end{aligned} \qquad \ldots\ldots (1)$$

$$\begin{aligned} Q &= C_4 d^3; \quad H = C_5 d^2; \\ P &= C_6 d^5; \quad E = K; \\ N &= K \end{aligned} \qquad \ldots\ldots (2)$$

$$\begin{aligned} Q &= C_7 N d^3; \quad H = C_8 N^2 d^2; \\ P &= C_9 N^3 d^5; \quad E = K \end{aligned} \qquad \ldots\ldots (3)$$

From Equation 3 for H, the following may be derived:

$$\sqrt{H} = C_{10} N d; \; C_{10} = \frac{\sqrt{H}}{Nd} \qquad \ldots (4)$$

Equation 3 for Q and Equation 4 for H may be combined as follows:

$$\frac{Q}{\sqrt{H}} = \frac{C_7 N d^3}{C_{10} N d} = C_{11} d^2;$$

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* Chief Engr., Division of Industrial Hygiene and Safety Standards, New York State Department of Labor.

** Industrial Hygiene Engr., Division of Industrial Hygiene and Safety Standards, New York State Department of Labor.

For presentation at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Bretton Woods, N. H., June 1948.

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$$C_{11} = \frac{Q}{d^2\sqrt{H}} \quad \ldots\ldots\ldots\ldots (5)$$

For any homologous series of fans, the fan inlet diameter D may be used instead of the fan wheel diameter d. Therefore, Equations 4 and 5 for $\frac{\sqrt{H}}{Nd}$ and $\frac{Q}{d^2\sqrt{H}}$ may be rewritten respectively as:

$$C_{12} = \frac{\sqrt{H}}{ND} \quad \ldots\ldots\ldots\ldots (6)$$

$$C_{13} = \frac{Q}{D^2\sqrt{H}} \quad \ldots\ldots\ldots\ldots (7)$$

In Equation 7, Q and D are related by the velocity V through the fan inlet in the following manner:

$$Q = \frac{\pi D^2 V}{4}; \quad Q = C_{14} D^2 V;$$

$$\frac{Q}{D^2} = C_{14} V \quad \ldots\ldots\ldots\ldots (8)$$

The expression for $\frac{Q}{D^2}$ developed in Equation 8 may be substituted in Equation 7 with the result:

$$C_{13} = \frac{C_{14} V}{\sqrt{H}}; \; C_{15} = \frac{V}{\sqrt{H}} \quad \ldots\ldots (9)$$

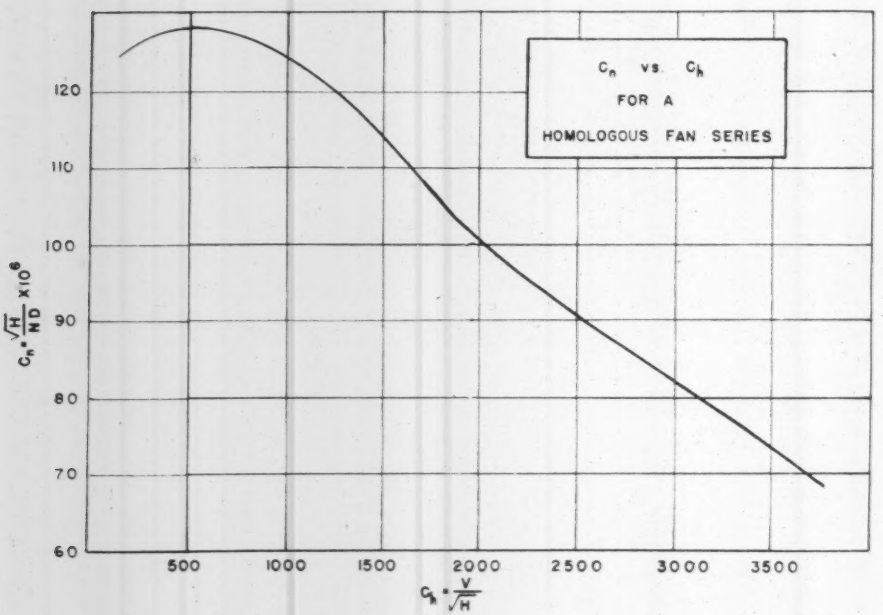

Fig. 1—Values of C_n *vs.* C_h for a homologous fan series

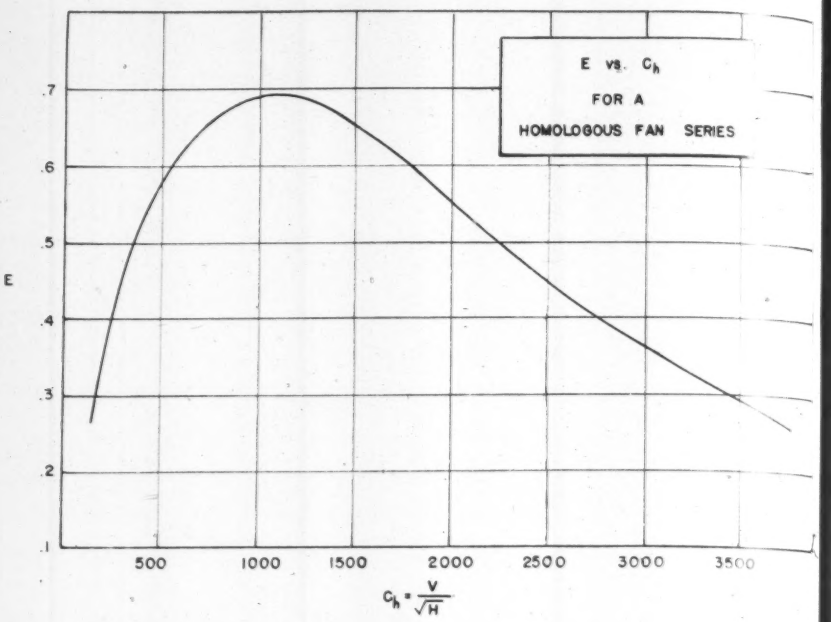

Fig. 2—Values of E *vs.* C_h for a homologous fan series

Constant C_{15} may be used to introduce static pressure into the solution of fan problems and may therefore be called, for convenience, the *Static Pressure Constant*—C_h. Similarly, constant C_{13} may be used to introduce fan speed into the solution of fan problems and, in like manner, may be called the *Speed Constant*—C_n. Thus:

$$C_h = \frac{V}{\sqrt{H}}; \; C_n = \frac{\sqrt{H}}{ND} \quad \ldots\ldots (10)$$

The reason for selecting this particular set of constants rather than any other combination which could have been chosen, is that in C_h and C_n, (1) N, D, and V appear to the first power, and (2) H appears in both constants as the same power. Because of this both C_h and C_n may easily be computed either by means of a simple slide rule or a table of square roots of the usual values of H.

Fan horsepower, P, is obtained from the equation:

$$P = \frac{0.0001575\ QH}{E} \quad \ldots\ldots (11)$$

In Equation 11, E is the ratio of theoretical air horsepower ($0.0001575\ QH$) to actual horsepower P.

In the usual fan selection problem, Q, D, H and V are known before the fan type, speed and horsepower are chosen. Since V and H are known, C_h which equals $\frac{V}{\sqrt{H}}$ also has a known value.

Similarly, since D and H are known, the product $N \times C_n$ which equals $\frac{\sqrt{H}}{D}$ also has a known value.

Finally, since Q and H are known and the value of the constant 0.0001575 is established, the product $P \times E$ which equals $0.0001575QH$ also has a known value. Therefore, to find N and P respectively, it is only necessary to provide a convenient means for finding C_n and E.

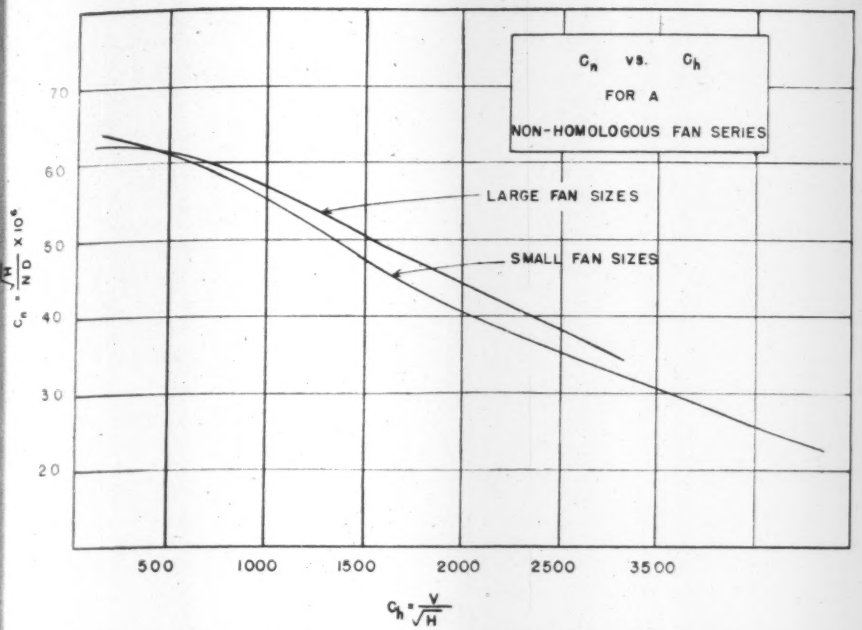

Fig. 3—Values of C_n *vs.* C_h for a nonhomologous fan series

It is a primary purpose of this paper to show that the desired convenient means is provided (1) by separately plotting C_n and E against C_h and, (2) separately tabulating values of C_n and E for selected values of C_h and obtaining intermediate values between those tabulated by interpolation.

A homologous series of fans is one in which performance follows Equation 2. Some fan manufacturers design all sizes of a particular fan type as a homologous series; others change the basic design one or more times in assembling the whole size range of the same nominal type of fan. Where the former is true, all sizes of the fan type can be represented by one C_h - C_n and one C_h - E curve or table (Figs. 1 and 2) (Table 1). Where the latter is the case, two or more curves or tables are required to represent all sizes of the fan type (Figs. 3 and 4) (Table 2).

An analysis of the C_h-C_n and C_n-E curves for 23 series of fans suppled for this study by eight of the principal fan manufacturers shows three distinct groups of C_h-C_n curves. Those for high speed backward curve fans have values of $C_n = \frac{\sqrt{H}}{ND} \times 10^6$ for maximum efficiency between 47 and 60; those for forward curve (squirrel cage) fans have values of the same constant for maximum efficiency between 85 and 93; and the paddle wheel, high pressure and special low speed wheel designs fall in the range from 100 to 145. Figs. 1 and 2 represent a typical fan in this latter category, whereas Figs. 3 and 4 represent a fan of the high speed backward curve type. The fact that Fig. 4 has a peak at a slightly higher efficiency than Fig. 2 does not imply that high speed fans are more efficient as a class than low speed fans, but merely that this occurred in the particular pair of fans selected for illustration in this paper.

It is convenient to use values of $\frac{\sqrt{H}}{ND} \times 10^6$ for purposes of plotting since this yields numbers greater than 1. Because of this, the process of calculating N from known values of V, H and D is as follows:

(*a*) Calculate $C_h = \frac{V}{\sqrt{H}}$

(*b*) Determine $C_n = \frac{\sqrt{H}}{ND} \times 10^6$ from C_h - C_n curve

(*c*) Note that $\frac{\sqrt{H}}{ND} = C_n \times 10^{-6}$ therefore:

(*d*) $N = \frac{\sqrt{H} \times 10^6}{C_n D}$

When Q rather than V is known, the relation between Q and V is:

$$Q = \frac{\pi D^2 V}{4 \times 144} = \frac{D^2 V}{183.3} \quad \ldots (12)$$

The process of calculating horsepower from known values of V, H and D is as follows:

(*a*) Calculate $C_h = \frac{V}{\sqrt{H}}$

(*b*) Determine E from C_h - E curve

(*c*) Calculate Q from Equation 12

(*d*) $P = \frac{0.0001575\, QH}{E}$

The use of Figs. 1 through 4 and Tables 1 and 2 in computing fan

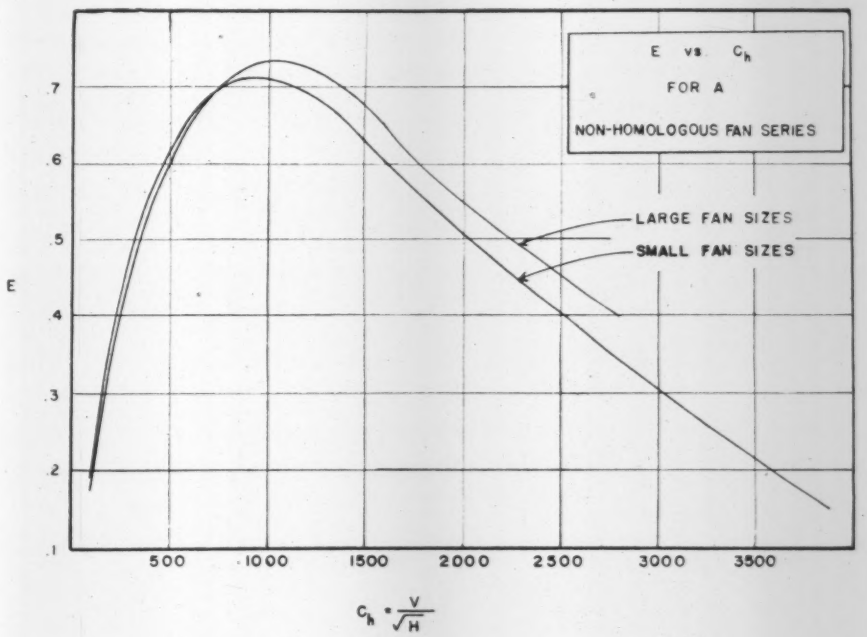

Fig. 4—Values of E *vs.* C_h for a nonhomologous fan series

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speed and horsepower are illustrated in the following examples:

Example 1. (Use of Fig. 1 or Table 1). Given: $V = 3500$ fpm, $D = 17$ in., $H = 6$ in. Find required fan speed, N.

Solution:

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{3500}{\sqrt{6}} = 1430$

(b) $C_n = \dfrac{ND}{D} \times 10^6 = 116.3$
(from Fig. 1 or Table 1)

(c) $\dfrac{\sqrt{H}}{ND} = C_n \times 10^{-6} = 116.3 \times 10^{-6}$

(d) $N = \dfrac{\sqrt{H} \times 10^6}{C_n D} = \dfrac{2.449 \times 10^6}{116.3 \times 17}$
$= 1240$ rpm
From catalog data $N = 1236$ rpm

Example 2. (Use of Fig. 2 or Table 1). Given: $V = 3500$ fpm, $D = 17$ in., $H = 6$ in. Find required fan horsepower input, P.

Solution:

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{3500}{\sqrt{6}} = 1430$

(b) $E = 0.663$ (from Fig. 2 or Table 1)

(c) $Q = \dfrac{D^2 V}{183.3} = \dfrac{17^2 \times 3500}{183.3}$
$= 5530$ fpm (from Equation 12)

(d) $P = \dfrac{0.0001575\, QH}{E}$
$= \dfrac{0.0001575 \times 5530 \times 6}{0.663}$
$= 7.85$ hp
From catalog data the horsepower input = 7.9

Example 3. (Use of Fig. 3 or Table 2). Given: $D = 15.75$ in., $A_o = 1.28$ sq ft, $V_o = 1800$ fpm, $H = 4$ in. Find required fan speed, N.

Solution: Since the inlet area is 1.34 sq ft,

$V = \dfrac{V_o A_o}{A} = 1800 \times \dfrac{1.28}{1.34}$
$= 1720$ fpm

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{1720}{\sqrt{4}} = 860$

(b) $C_n = \dfrac{\sqrt{H}}{ND} \times 10^6 = 57$
(From Fig. 3 or Table 2)

(c) $\dfrac{\sqrt{H}}{ND} = C_n \times 10^{-6} = 57 \times 10^{-6}$

(d) $N = \dfrac{\sqrt{H} \times 10^6}{C_n D} = \dfrac{2 \times 10^6}{57 \times 15.75}$
$= 2230$ rpm
From catalog data the required rpm = 2264

Example 4. (Use of Fig. 4 or Table 2). Given: $D = 15.75$ in., $V = 1720$ fpm, $H = 4$ in. Find required horsepower input P.

Solution:

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{1720}{\sqrt{4}} = 860$

(b) $E = 0.710$
(From Fig. 4 or Table 2)

(c) $Q = \dfrac{D^2 V}{183.3} = \dfrac{15.75^2 \times 1720}{183.3}$
$= 2300$ fpm
(From Equation 12)

(d) $P = \dfrac{0.0001575\, QH}{E}$
$= \dfrac{0.0001575 \times 2300 \times 4}{0.710}$
$= 2.04$ hp
From catalog data the horsepower input = 2.05

Example 5. (Use of Fig. 3 or Table 2). Given: $D = 28$ in., $V = 3000$ cfm, $A_o = 4.18$ sq ft, $A = 4.28$ sq ft, $H = 2.0$ in. Find the required fan speed, N.

Solution: Since inlet and outlet are known

$V = 3000 \times \dfrac{4.18}{4.28} = 2940$ fpm

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{2940}{\sqrt{2}} = 2080$

(b) $C_n = \dfrac{\sqrt{H}}{ND} \times 10^6 = 43.5$
(From Fig. 3 or Table 2)

(c) $\dfrac{\sqrt{H}}{ND} = C_n \times 10^{-6} = 43.5 \times 10^{-6}$

(d) $N = \dfrac{\sqrt{H} \times 10^6}{C_n D} = \dfrac{1.44 \times 10^6}{43.5 \times 28}$
$= 1160$ rpm
From catalog data the required rpm = 1231

Example 6. (Use of Fig. 4 or Table 2). Given: $D = 28$ in., $V = 2940$ fpm, $H = 2$. Find the required horsepower input, P.

Solution:

(a) $C_h = \dfrac{V}{\sqrt{H}} = \dfrac{2940}{\sqrt{2}} = 2080$

(b) $E = 0.53$
(From Fig. 4 or Table 2

Table 1—Data Representing Homologous Fan Series

$C_h = \frac{V}{\sqrt{H}}$	250	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000	3250	3500
$C_n = \frac{\sqrt{H}}{ND} \times 10^6$ ...	126.5	128.4	127.5	125.0	120.5	114.3	107.0	100.5	95.20	90.50	86.00	81.50	77.50	73.50
Efficiency	0.400	0.525	0.653	0.687	0.686	0.652	0.605	0.550	0.495	0.445	0.400	0.360	0.330	0.295

Table 2—Data Representing Nonhomologous Fan Series Comprising Two Homologous Size Ranges

$C_h = \frac{V}{\sqrt{H}}$	250	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000	3250	3500
	Small Fan Sizes (7 Fan Sizes in Series)													
$C_n = \frac{\sqrt{H}}{ND} \times 10^6$ ..	63.0	61.0	58.5	55.0	51.3	47.5	43.5	40.3	37.5	35.0	32.5	30.5	28.3	26.5
Efficiency	0.435	0.620	0.700	0.711	0.685	0.630	0.567	0.510	0.455	0.400	0.355	0.310	0.285	0.263
	Large Fan Sizes (16 Fan Sizes in Series)													
$C_n = \frac{\sqrt{H}}{ND} \times 10^6$ ..	65.0	61.5	59.5	57.0	53.7	50.0	47.5	44.5	41.5	38.0	35.5	—	—	—
Efficiency	0.410	0.605	0.705	0.735	0.720	0.675	0.610	0.55	0.500	0.455	0.407	—	—	—

Table 3—Value of H and $\sqrt{H}$

H	$\sqrt{H}$	H	$\sqrt{H}$	H	$\sqrt{H}$	H	$\sqrt{H}$
¼	0.500	2¼	1.500	5	2.236	9½	3.082
⅜	0.622	2½	1.581	5¼	2.290	10	3.162
½	0.708	2¾	1.660	5½	2.345	11	3.370
⅝	0.792	3	1.732	5¾	2.400	12	3.464
¾	0.866	3¼	1.810	6	2.449	13	3.606
⅞	0.935	3½	1.871	6½	2.550	14	3.742
1	1.000	3¾	1.940	7	2.646	15	3.873
1¼	1.120	4	2.000	7½	2.739	16	4.000
1½	1.325	4½	2.120	8½	2.915	..	....
2	1.414	4¾	2.180	9	3.000	..	....

(c) $$Q = \frac{D^2 V}{183.3} = \frac{28^2 \times 2940}{183.3}$$

$$= 12{,}600 \text{ cfm}$$

(From Equation 12)

(d) $$P = \frac{0.0001575\, QH}{E}$$

$$= \frac{0.0001575 \times 2 \times 12600}{0.53}$$

$$= 7.45 \text{ hp}$$

From catalog data the required horsepower input = 7.30.

Advantages of Proposed Method

Information usually found in a catalog fan rating table of six or more pages is condensed into a single line of data in a table listing the ratings of a number of different fans. For *office use,* the detailed manufacturers' multi-rating table, several pages in length, is more convenient than the method proposed in this paper. However, for *field use,* this method substitutes one printed table in a notebook or briefcase for many pounds of catalogs and multi-rating tables which would otherwise have to be transported to provide ready reference to equivalent information. If a table of square root values of H as commonly listed in fan multi-rating tables is available (Table 3), the computation resolves itself to first power multiplication and division and can therefore be made without a slide rule. The fact that the use of this method requires interpolation necessarily decreases its accuracy. However this objection is more academic than real because, (1) the multi-rating tables themselves require interpolation and (2) once the correct value of speed and horsepower is determined from either a multi-rating table or by means of this method the fan selected will actually be driven by a motor or motor and pulley combination available in a limited number of speeds and horsepowers. Thus, if the correct speed and horsepower are 1480 rpm and 2.72 hp, the fan would probably be belt driven by a 3 hp-1725 rpm motor because the pulley ratio to yield exactly 1480 rpm from a 1725 rpm motor is not readily obtainable. The nearest rpm values actually obtainable are 1487 and 1474 respectively.

By using a slide rule, fan speed can, in almost all cases, be predicted by this method to within less than 2 percent of values published in fan rating tables. However, in the example just cited where 1480 rpm is desired there is actually a range of 12 rpm between the two nearest available speeds. Similarly in the typical multi-rating table, the difference between successive listed speeds is generally over 10 rpm for constant static pressure, and over 100 rpm for constant capacity. It is felt, therefore, that in practice there is not much difference in the accuracy of cross-interpolation in a multi-rating table and that resulting from single interpolation and slide rule computation according to the proposed field method.

The method proposed will yield horsepower values accurate to the first decimal place in the range of motors made in half-horsepower sizes and to at least the whole number in the range beyond half-horsepower sizes. In practice, therefore, it will allow the selection of motor size to the same degree of accuracy as could be obtained through the use of a multi-rating table.

The proposed method is particularly valuable in the correction in the field of ventilation systems where the fan is moving too little or occasionally too much air. Here it is easy to calculate what should be the new capacity Q and the corresponding inlet velocity V for the existing fan inlet duct. Static pressure H can be calculated from an inspection of the existing system. From this information C_h is obtainable. Reference to a convenient table of values of C_n and E for various makes of fans allows the rapid calculation of the new fan speed and horsepower needed to make the system perform properly. Since, in the field, men would never have a full set of fan multi-rating tables available, such calculation would be impossible to accomplish in the field by any heretofore available methods of fan rating.

In general, industrial exhaust systems operate with fan inlet velocities between 1500 and 4500 fpm. Velocities close to 2000 fpm are used for gas and fume exhaust systems where it is unnecessary to maintain dust carrying velocities. For this application velocities range from about 1500 to 2500 fpm and fan static pressures are generally between ½ in. and 2½ in. For dust transport, velocities close to 4000 fpm are employed, the range being from about 3000 to 4500 fpm. Static pressures for this type of operation generally range from 4 in. to 7 in. Therefore C_h for dust carrying systems varies from 1125 to 2250, and for non-dust carrying systems from 950 to a practical maximum of about 2200. Figs. 2 and 4 show that maximum fan efficiency usually occurs at values of C_h close to or below 1000. In other words these figures show quite strikingly the extent to which ordinary industrial exhaust systems operate at relatively inefficient portions of the fan characteristic curves.

The fan constants C_h and C_n offer tempting possibilities for further analysis. By grouping fans by blade type and number of blades, and further, by arranging the grouped data in terms of d/D and D/w ratios, a number of interesting generalities of fan performance become evident. For instance, there is a strong correlation between d/D ratios for conventionally designed six or eight blade paddle wheel fans of low D/w ratio (*i.e.* less than 1.3) and the value of C_n for any constant value of C_h. This fact allows interpolation within reasonable limits between the tabular data of the various manufacturers of this very common type of fan.

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ASHVE RESEARCH PROGRAM

Studies under the direction of the

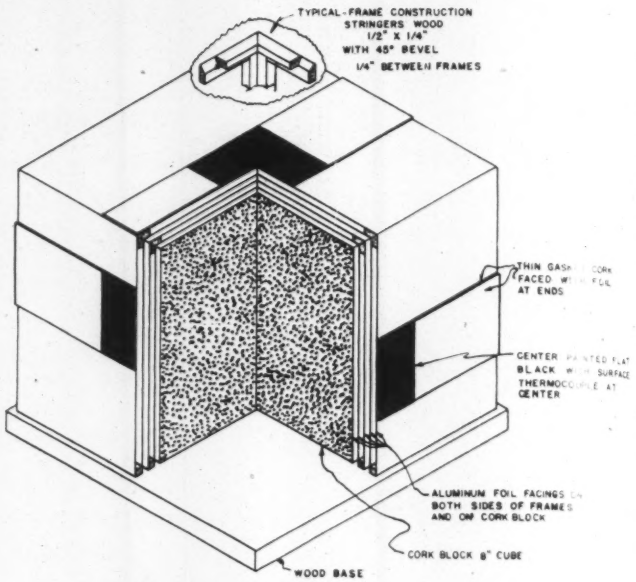
TECHNICAL ADVISORY COMMITTEE on COOLING LOAD

Scope

The evaluation of all factors involved in calculations of cooling load for comfort and process air conditioning.

Program Completed 1946-47

1. The development of the sol-air temperature concept as a means of evaluating the combined effect of air temperature plus sun radiation and sky radiation. The design and construction of a sol-air thermometer. The calculation of sol-air temperatures for two cities typical of industrial and rural atmospheres. (Three research papers published in cooperation with Cornell University.)
2. The analysis of periodic heat flow through a homogeneous wall or roof and through composite walls and roofs. (Three papers published in cooperation with Cornell University and the University of California.)
3. Compilation of data on properties of common building materials used in periodic and steady state heat flow calculations. (ASHVE Research Bulletin, Vol. 53, No. 2.)



SOL-AIR TEMPERATURE METER

Construction Details of Sol-Air Thermometer

Program in Progress and Planned for 1948

1. The calculation of sol-air temperature data for a humid-area atmosphere (New Orleans, La.).
2. A study of the effect of size on the characteristics of the sol-air thermometer.
3. The development of tables for determining the instantaneous rates of heat flow through various solid structures.
4. A study of the absorptivity of various paints for solar radiation.
5. The investigation of the effects on cooling loads of the heat gains due to appliances and lighting fixtures.

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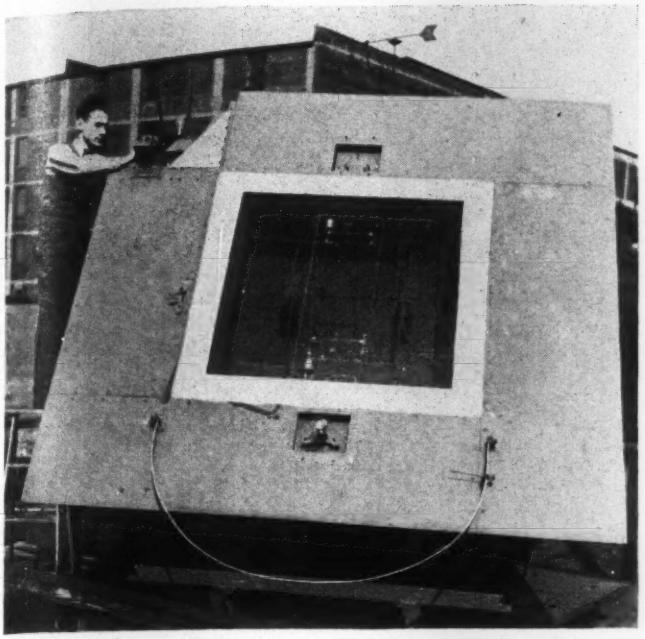

Unique type of calorimeter designed by the Laboratory staff to measure solar energy transmission through glass. It will accommodate sheet, plate, and configurated glass, glass block and shading devices.

Studies under the direction of the TECHNICAL ADVISORY COMMITTEE on HEAT FLOW THROUGH GLASS

Scope

The investigation of heat flow through glass areas, whether by direct transmission or absorption and re-radiation of radiant energy, or by conduction and convection. The significance of these effects in problems of space cooling and heating loads.

Program Completed 1946-47

1. Review and critical analysis of published and unpublished data on heat transmission through glass. (ASHVE Research Bulletin, Vol. 53, No. 1, on the press.)
2. Studies of forced convection heat transfer from flat surfaces. (Two research papers and ASHVE Research Bulletin, Vol. 53, No. 3, published.)
3. Experimental and analytical studies of transmittance of direct solar radiation by five different types of glass in single sheets and four combinations of two sheets of glass spaced 1/4 in. apart. (One research paper published, one on the press.)
4. Determination of overall coefficients of heat transfer for single and double glass.

Program in Progress 1948

1. Investigation of solar and total heat transmittance through eight glass block panels in five different patterns and three different sizes.
2. Studies of radiation exchange between glass areas and their surroundings, uninfluenced by solar radiation.
3. Studies of heat transfer from a smooth flat plate to low velocity air streams.
4. Investigation of the effects of solar energy distribution.

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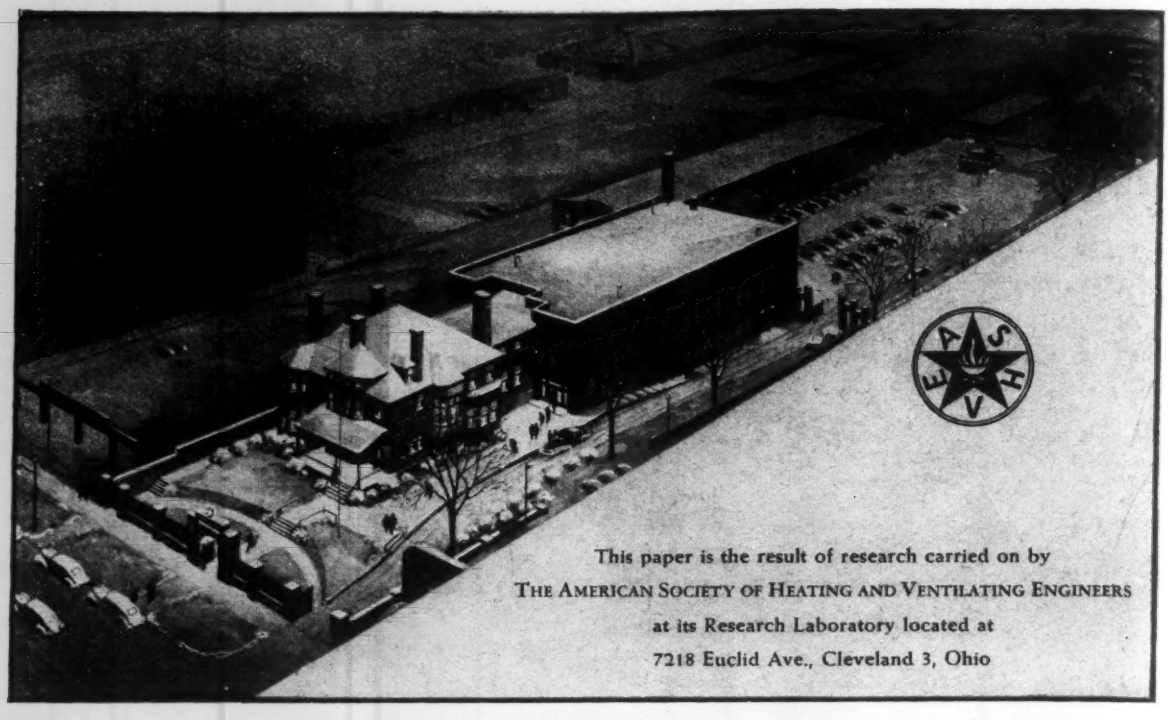
This paper is the result of research carried on by
THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
at its Research Laboratory located at
7218 Euclid Ave., Cleveland 3, Ohio

Solar and Total Heat Gain Through Double Flat Glass

By George V. Parmelee* and Warren W. Aubele**, Cleveland, Ohio

SUMMARY—A report of tests conducted at the ASHVE Research Laboratory to determine solar transmittance of various combinations of two sheets of flat glass unshaded and spaced ¼ in. apart.

Transmittance data for direct and diffuse solar radiation are given. Effects of solar energy distribution are shown. The influence on heat gain of direct transmission and of re-radiation and convection from the glass is shown by examples of total heat gain through windows with a western exposure.

Introduction

THIS paper presents results of studies of solar heat gain through flat glass and is essentially a second progress report of an investigation being carried on under the direction of the Committee on Research and the Technical Advisory Committee on Glass†. The data deal primarily with the transmittances of various combinations of two sheets of glass, unshaded and spaced ¼ in. apart. Data are also presented on rates of heat transfer from the indoor sheet of glass to the calorimeter apparatus which has been used in these studies.

An earlier paper[1] described the apparatus and methods of conducting the tests. Data were presented for several types of single glass and showed the relationship between angle of the incident ray of sunlight and transmittance. Values of diffuse solar radiation transmittance were also given. The effect of the type of glass on total heat gain was illustrated and the significance of solar energy distribution was pointed out.

The results which appear in this paper are subject to the following limitations:

1. Uncertainty regarding the exact energy distribution of the direct and diffuse solar radiation, and
2. Incomplete data regarding the heat transfer processes which take place at the outdoor surface of the window.

Research is in progress to seek answers to these problems as well as to study other phases of heat gain through windows and glass block panels. Therefore, no attempt has been made to present a complete interpretation of the test data.

Apparatus

The Calorimeter

A detailed description of the calorimeter apparatus has already been given in a previous report[1] and will not be repeated here. Fig. 1 shows some features of the apparatus, which consists primarily of a calorimeter fastened to the rear side of a window frame which can be rotated to any point of the compass and tilted through 90 deg. The

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* Research fellow, ASHVE Research Laboratory. Member of ASHVE.

** Assistant research engineer, ASHVE Research Laboratory.

† Committee Personnel: R. A. Miller, *Chairman*; A. B. Algren, F. L. Bishop, Jr., J. E. Frazier, J. S. Herbert, E. H. Hobbie, L. K. Jones, C. O. Mackey, F. W. Preston, W. C. Randall, Vic Sanders, H. B. Vincent, G. B. Watkins, F. C. Wood.

[1] Exponent numerals refer to Bibliography.

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Bretton Woods, N. H., June 1948.

calorimeter is heavily insulated and consists of a blackened sheet aluminum shell to which is brazed a grid of tubes. A mixture of ethylene-glycol and water is circulated through these tubes. The glass sheets are installed in a clear opening 44½ in. square. Although the outer sheet of glass in the double glass installation is flush with the front face of the window frame, the inner sheet is shaded by virtue of its set-back to an extent dependent upon the incident angle, the solar altitude and the spacing of the two sheets.

The apparatus is used in two ways. Fixed angle tests are conducted by setting the sun sight shown in the upper left of Fig. 1 at the desired angle and rotating and tilting the apparatus so that the sun's rays always pass through the tube. Other tests are run with the glass vertical and in a fixed position. In such instances the sun sight is used to measure the incident angle.

Instrumentation

Thermocouples are used to measure the following temperatures: calorimeter surface (ten points), calorimeter air (four points), glass surfaces (four to eight points), and outdoor air. The temperature rise of the coolant is measured by a multi-junction thermopile which is connected to five points of a sixteen-point recording potentiometer. The actual coolant temperature, both inlet and outlet, is measured by separate thermocouples.

The flow rate of the coolant is measured by calibrated thin plate orifices and a mercury manometer.

Measurements of total solar irradiation are made by Eppley thermo-electric pyrheliometers. At least one is mounted on the face of the window frame (see Fig. 1). A second is mounted on a horizontal plane on the weather instrument tower. These instruments are connected to the recording potentiometer. By locally shading the thermo-electric element, measurements of diffuse solar radiation are also made. The difference between the total and diffuse radiation is the direct or beamed radiation.

In addition to these measurements, the wind velocity and direction and the sky condition are noted. Wet and dry bulb temperatures of the air are also obtained.

Significant terms in this report are defined as follows:

Definitions

Direct solar radiation.....J_D = radiation from the sun in a beam of parallel rays.

Direct normal solar radiation.J_{Dn} = radiation from the sun measured on a plane normal to the sun's rays.

Diffuse solar radiation.....J_d = solar radiation received from the sky by reason of the sun's rays being scattered and reflected by the atmospheric constituents; also refers to solar radiation reflected diffusely from ground objects.

Total incident solar radiation..J_T = direct plus diffuse solar radiation measured in the plane of a window.

Solar altitude.....β_a = the angular position of the sun above the horizon.

Air mass.....M = relative path length of the sun's rays through the atmosphere; for example, air mass 2 means that the sun's position is such that its path length is double that when it is directly overhead.

Angle of incidence.....θ = angle between the sun's rays and a perpendicular to the window.

Atmospheric transmittance.....τ_a = the direct normal solar radiation at the earth's surface divided by the solar constant.

Solar constant.....J_c = the intensity of solar radiation on a plane perpendicular to the rays outside the earth's atmosphere.

Transmitted solar energy.....J_τ = that portion of the incident solar radiation which passes directly through glass as radiation and which has the same wave length spectrum as the incident radiation.

Transmittance .....τ = the property of glass for allowing solar radiation to pass through expressed as a percentage or a fraction. Subscripts $_D$, $_d$, $_n$, and $_T$ refer to this property for direct, diffuse, normally incident, and total radiation.

Reflectance .....ρ = the property of glass for reflecting solar energy expressed as a percentage or a fraction. Subscripts $_D$, $_d$, $_n$, and $_T$ refer to this property for direct, diffuse, normally incident, and total radiation.

Absorptance .....α = the property of glass for absorbing solar energy expressed as a percentage or a fraction. Subscripts $_D$, $_d$, $_n$, and $_T$ refer to this property for direct, diffuse, normally incident, and total radiation.

Absorption characteristic.....KL = a dimensionless property of glass, where K is the absorption coefficient in units per inch, and L is glass thickness in inches.

Fresnel reflectance.....r = the reflectance at a single glass-air interface (note: this depends upon the plane of polarization and θ).

Computations

Solar Heat Gain

The solar energy which strikes a window or glass block panel comes principally from the direct rays of the sun. Significant amounts, usually in the form of diffusely reflected radiation, may, however, come from other sources. Fig. 2 illustrates common sources. The figure also gives approximate ranges of intensity of the radiation from the sun and the sky as well as the breadth of the spectral bands in which the energy from these two sources is concentrated.

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The solar energy which is transmitted by glass from all sources equals the incident radiation minus that absorbed minus that reflected. Expressed in equation form,

$$J_T \tau_T = J_T - J_T \rho_T - J_T \alpha_T \ldots (1)$$

where

J_T = total incident radiation, Btu per (hr) (sq ft)

α, ρ, τ = absorptance, reflectance, and transmittance for the total incident radiation.

Most of that which is transmitted through the glass is absorbed by the indoor surfaces of the enclosure. Some radiation may be reflected back through a window by shades or other reflecting surfaces.

That portion of the incident solar radiation which is absorbed by the glass results in a rise in temperature of the glass. At any instant part of this is stored and the rest is dissipated by convection and radiation from the indoor and outdoor surfaces. In equation form this heat balance is:

$$J_T \alpha_T = h_o (t_{go} - t_o) + R_o + h_i (t_{gi} - t_i) + R_i + S \ldots (2)$$

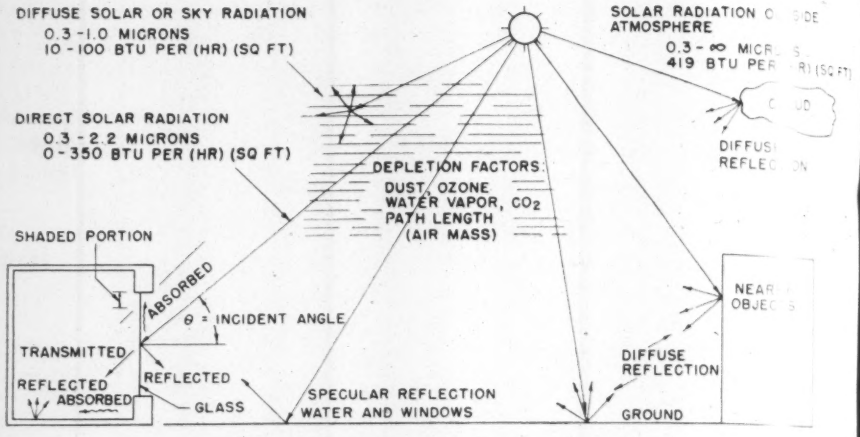

Fig. 2—Sources of solar radiation

where

h_o, h_i = the outdoor and indoor convection coefficients, Btu per (hr) (sq ft) (F deg)

t_{go}, t_{gi} = the outdoor and indoor glass surface temperatures

t_o, t_i = the outdoor and indoor air temperatures

R_o, R_i = the outdoor and indoor net radiation exchanges between the outdoor and indoor glass surfaces and their respective surroundings, Btu per (hr) (sq ft)

S = the rate of heat storage in the glass, Btu per (hr) (sq ft).

Fig. 3 illustrates these heat transfer and exchange processes. Although glass is selectively transparent to radiation between wave lengths of about 0.3 and 4.75 microns, beyond this point it is opaque and behaves approximately as a black body to radiation from objects at temperatures up to several hundred Fahrenheit degrees. The figure illustrates the complex nature of the radiant heat exchange between an outdoor glass surface and ground objects, clouds, and the atmosphere. The last selectively absorbs and emits in wave length bands approximately as indicated. The radiation exchange on the indoor side may be equally complex.

From test data obtained by means of the calorimeter apparatus the following quantities are determined:

1. Heat gain by the coolant (by integration of the potentiometer record).
2. Heat transferred by radiation and convection to the inner surface of the inner glass from the calorimeter (computed),
3. Heat lost through the calorimeter insulation (determined by calibration),
4. Heat lost through the edge or corner block holding the glass (determined by calibration),
5. Heat stored in the calorimeter shell by virtue of a rise in its temperature (computed).

The total transmitted solar energy, $J_T \tau_T$, is determined by finding the algebraic sum, q_s, of these quantities. Thus:

$$J_T \tau_T = q_s \ldots (3)$$

Transmittances

The transmittance of the glass for diffuse solar radiation, τ_d, is found by orienting the calorimeter apparatus so that no direct sun-

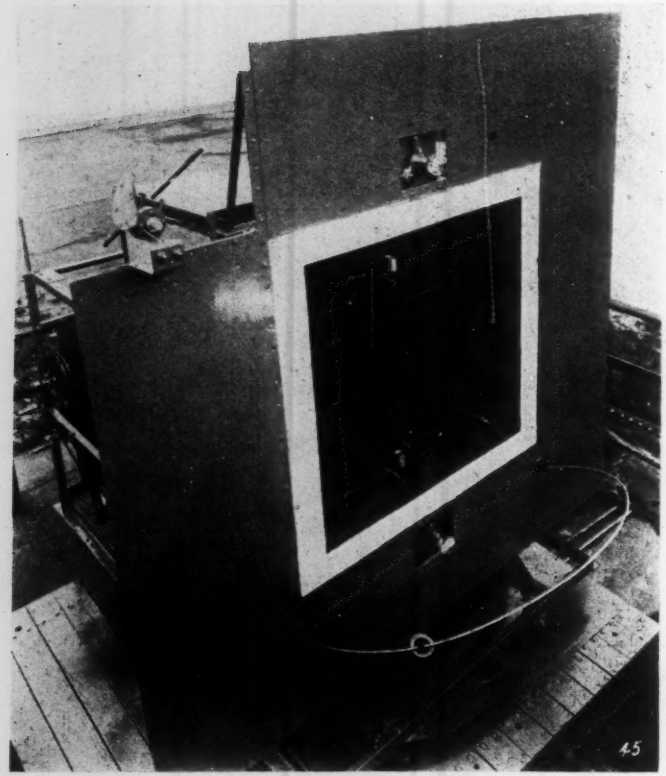

Fig. 1—Close up view of calorimeter in location for test

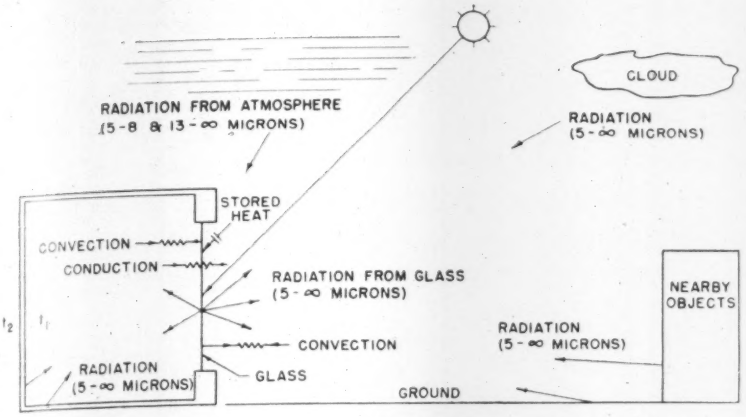

Fig. 3—Convection heat transfer and radiation exchanges at glass surfaces

light strikes the glass. The transmittance is then:

$$\tau_d = q_s/J_d \quad \ldots\ldots (4)$$

where

q_s = transmitted solar radiation, Btu per (hr) (sq ft) gained by the calorimeter and computed as just outlined

J_d = incident diffuse radiation, Btu per (hr) (sq ft) computed from the integrated record of the pyrheliometer emf.

The diffuse transmittance may also be found, as described later, by means of the Eppley pyrheliometer.

The transmittance of direct radiation, τ_D, is found by exposing the window to direct sunlight at the desired angle and is

$$\tau_D = \frac{q_s - J_d\tau_d}{J_T - J_d} = \frac{q_s - J_d\tau_d}{J_D} \quad \ldots (5)$$

where

J_T = total incident radiation, Btu per (hr) (sq ft) computed from the integrated record of the pyrheliometer emf

J_D = incident direct radiation, Btu per (hr) (sq ft).

The direct radiation must in some cases be modified to take into account partial shading of the inner sheet. Partial shading from diffuse solar radiation is considered negligible in this installation.

It should be noted that the transmitted solar radiation does not include that part of the solar heat absorbed by the glass which passes to the calorimeter by radiation and convection. Nor does it include heat flow due to differences between the indoor and outdoor temperatures. These two quantities appear together as $h_1\ (t_{g1} - t_1) + R_1$, are referred to as *heat flow from glass to calorimeter*, and may be either positive or negative.

Results Obtained

Spectral Transmittance Characteristics

The spectral transmittance curves of the glasses for which data are given in this paper and for which data have been given previously are shown in Figs. 4 and 5. This information was obtained by spectrophotometer tests of samples taken from the same sheet from which the full-sized samples were taken. Glasses Nos. 1, 2, 6, and 10 are ¼ in. thick polished plate glass. Glass No. 3 is a rolled wire glass polished on both sides and is 0.29 in. thick. Glass No. 9 is a drawn double strength window glass 0.125 in. thick. Glasses 3, 6, and 10 are commonly termed heat-absorbing. As shown by the curves, they absorb strongly in the infrared region (0.75 to ∞ microns) without unduly reducing the transmittances in the visible region (0.40 to 0.75 microns). This property is produced by minute amounts of iron in the form of ferrous oxide. All glasses are relatively poor transmitters of ultraviolet radiation, but since the energy contained in this region (0.29 to 0.40 microns) is only about 5 percent of the total energy it is not important from the standpoint of solar heat gain through glass. All of these glasses are classed as soda-lime-silica glasses.

Transmittance From Calorimeter Tests

Fig. 6 shows the transmittance of four combinations of two sheets of glass spaced ¼ in. apart.

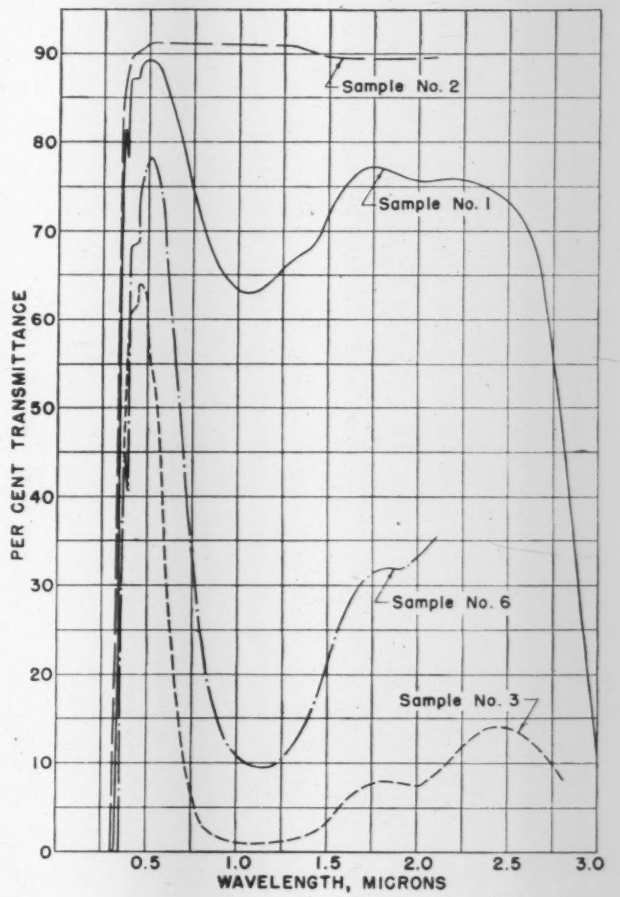

Fig. 4 — Spectral transmittance curves of regular and heat absorbing glasses

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These determinations were made by fixed angle tests in which the coolant was circulated through the calorimeter at a temperature which would result in the minimum difference between the indoor and outdoor temperatures. Under such a condition the corrections for heat gains or losses other than those due to directly transmitted solar radiation are at a minimum. There is, of course, a significant amount of heat transfer from the inner surface of the inner glass to the calorimeter if either sheet of glass absorbs appreciable amounts of solar radiation. This quantity is computed from the known temperatures of the glass and the calorimeter and from the heat transfer constants which have been separately determined for this apparatus. Because the inner sheet is partially shaded at high incident angles due to its set-back, appropriate corrections have been applied.

The values given in the curves apply, therefore, to sheets of glass large enough so that the partial shading, which is inevitable with any multiple glass installation, can be considered insignificant. It should be noted that these values apply to direct radiation received from the sun and measured *in a plane parallel to the window*. (For practical purposes it is sometimes convenient to use radiation intensities measured on a plane *perpendicular* to the sun's rays and to multiply these values by the cosine of the incident angle.)

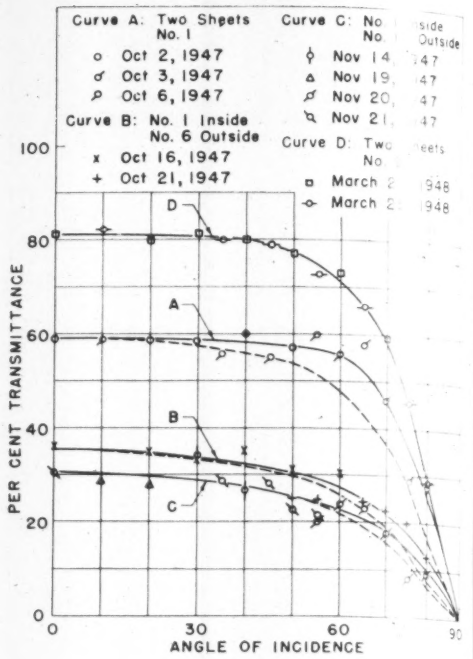

Fig. 5—Spectral transmittance curves of regular and heat absorbing glasses

Atmospheric conditions are known to affect transmittances because of their effect on solar energy distribution. It is, therefore, of interest to note the range of conditions under which the tests described here were conducted.

Transmittance Tests by Pyrheliometer

In addition to tests made by means of the calorimeter, normal incidence tests have been made by means of the Smithsonian silver disk pyrheliometer on both single glass and various combinations of double glass. The method of testing has been described in a previous report[1]. A considerable number of tests have been made on single glass, particularly No. 1, but only a few have been made on double glass. Because the test method requires that the solar intensity be very steady, the number of satis-

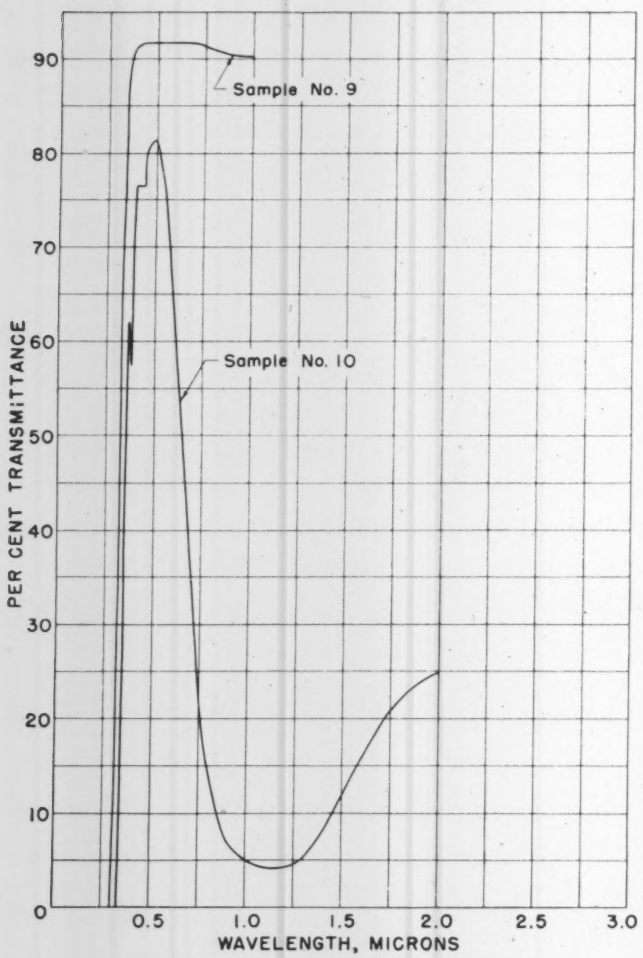

FIGURE SPECTRAL TRANSMITTANCE CURVES OF REGULAR AND HEAT ABSORBING GLASSES

Fig. 6—Transmittance of double glass for direct solar radiation

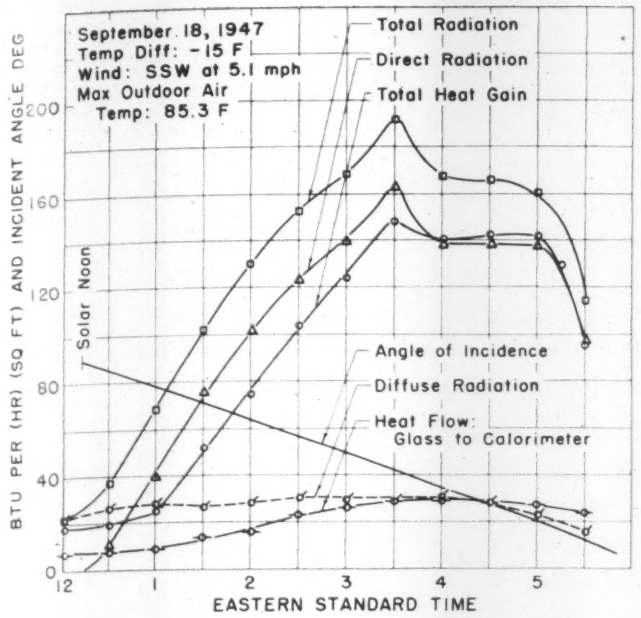

Fig. 7—Test results for double glass oriented west; both sheets glass No. 1; ¼ in. air space

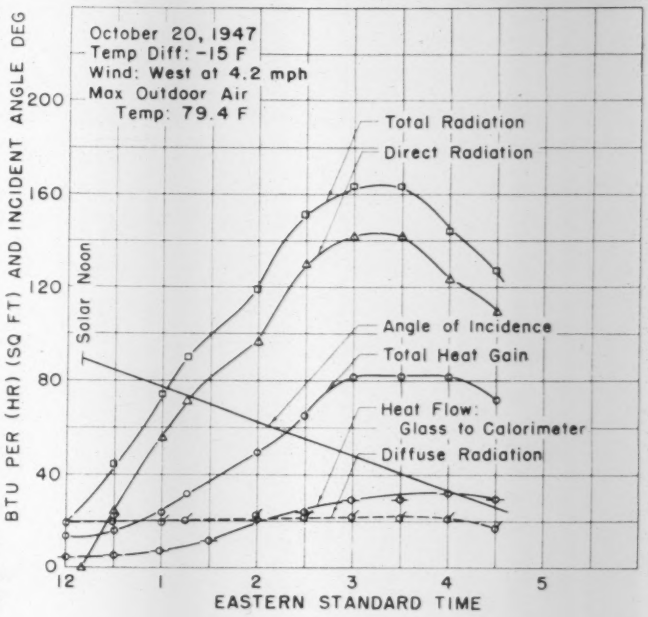

Fig. 8 *(right)*—Test results for double glass oriented west; inside sheet No. 1; outside sheet No. 6; ¼ in. air space

factory days that occur in a given period of time is rather limited. The results appear in Table 2.

The transmittances of various glasses for diffuse solar radiation have been measured by the Eppley pyrheliometer using four-foot square sheets. Because the transmittance of the cover glass of the Eppley pyrheliometer is somewhat altered when a sheet of glass is placed between it and the source of radiation, an allowance has been made for the partially polarized radiation which reaches it as a result of first passing through one or more sheets of glass being tested. The correction factor has been estimated at 0.99 for one sheet and 0.97 for two sheets.

Table 1—Test Conditions for Transmittances Given in Fig. 6

Combination	Solar Altitudes Degrees	Air Mass	Dew Point[a] F	Atmospheric Transmittance
No. 1 Inside—No. 1 Outside	15—45	1.4—3.8	40—55	0.25—0.50
No. 1 Inside—No. 6 Outside	12—40	1.5—5.0	50—60	0.20—0.50
No. 1 Inside—No. 10 Outside	8—30	2.0—6.0	25—30	0.05—0.45
No. 9 Inside—No. 9 Outside	23—50	1.3—2.6	25—40	0.35—0.70

[a] Of the atmosphere at ground level

Table 2—Transmittance of Several Glasses for Direct Normal and for Diffuse Solar Energy as Determined by Pyrheliometer Tests

Glass Combination	Normal Incidence Transmittance		Diffuse Transmittance Measured
	By Pyrheliometer	Computed[a]	
Single No. 9	0.87—0.92	0.91[b]	0.81—0.83
Single No. 1	0.71—0.78	0.77	0.64—0.70
Single No. 6	0.37—0.42	0.42	0.37—0.43
Single No. 10	0.33—0.37	0.38	0.34—0.46
No. 9 Inside—No. 9 Outside	0.78—0.79	0.84[b]	0.69—0.72
No. 1 Inside—No. 1 Outside	0.55—0.61	0.61	0.50—0.56
No. 1 Inside—No. 6 Outside	0.29—0.33	0.35	0.21—0.25
No. 1 Inside—No. 10 Outside	0.30—0.31	0.32	0.28—0.33
No. 6 Inside—No. 6 Outside	0.22	0.24	———
No. 10 Inside—No. 10 Outside	0.21—0.24	0.23	0.21—0.31

[a] Based upon Moon's[2] energy distribution for air mass = 2; ordinates at 0.05 micron intervals
[b] Over the range 0.30 to 1.00 microns.

The present state of knowledge concerning diffuse or sky radiation does not permit making allowance for two other factors in these tests. The first factor is that the diffuse radiation is known to be unequally polarized. This condition varies from point to point in the sky, reaching a maximum at about 90 deg from the sun. The second factor is that the diffuse radiation contains ultraviolet radiation to the extent of 15 to 20 percent on a very clear day. Coupled with this is the fact that the pyrheliometer cover glass does not transmit ultraviolet radiation nearly so well as it does infrared or visible radiation. The cover glass transmittance is therefore somewhat increased by virtue of the alteration in energy distribution caused by interposing a sheet or sheets of glass between the instrument and the sky. The results obtained, subject to the limitations just mentioned, also appear in Table 2.

It will be noted that a range of values has been assigned to both the direct normal and the diffuse transmittances for each glass. This has been done because transmittances are affected by variations in the energy distribution of the incident radiation due to atmospheric conditions. These tests are being

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continued in order that relationships may be developed between transmittance and the various factors which affect energy distribution, namely, dust, dewpoint (water vapor content), and air mass (path length).

Total Heat Gain

In addition to the fixed angle tests which were made to measure transmittance and heat gain under substantially steady state conditions, tests have been made with the glass set vertical and maintained at a fixed orientation. In these tests the effect of temperature difference between the indoors (the calorimeter) and the outdoors has been investigated. These tests also serve to bring out storage effects that might exist and show the influence of heat transfer conditions that regulate the dissipation of solar heat absorbed by the glass.

From the various tests conducted two have been selected to compare two different combinations under approximately the same conditions. The results are shown in Figs. 7 and 8. Fig. 7 is for two sheets of glass No. 1 spaced ¼ in. apart. There were no clouds of sufficient thickness to obscure the sun at any time, but shortly after the noon hour thin cirrus or cirro stratus clouds developed. This development was followed by a general hazy condition. The changeable nature of this atmospheric condition resulted in a somewhat ragged solar radiation curve. This is presented in Fig. 7 as smoothed curves of direct and total radiation. The diffuse radiation curve is also shown. Corrections not exceeding 3 Btu per (hr) (sq ft) have been made in the heat gain data wherever the calorimeter temperature was greater than or less than 15 deg below the outside air temperature. This correction is based upon an overall heat transfer coefficient U of 0.60, a value which night tests have shown to apply very closely to conditions existing in this test. Preliminary analysis of these data as well as a previously published theoretical study[3] indicates that small corrections for temperature differences can, without significant error, be superimposed upon heat gain due to the combined effects of absorbed solar heat and temperature difference.

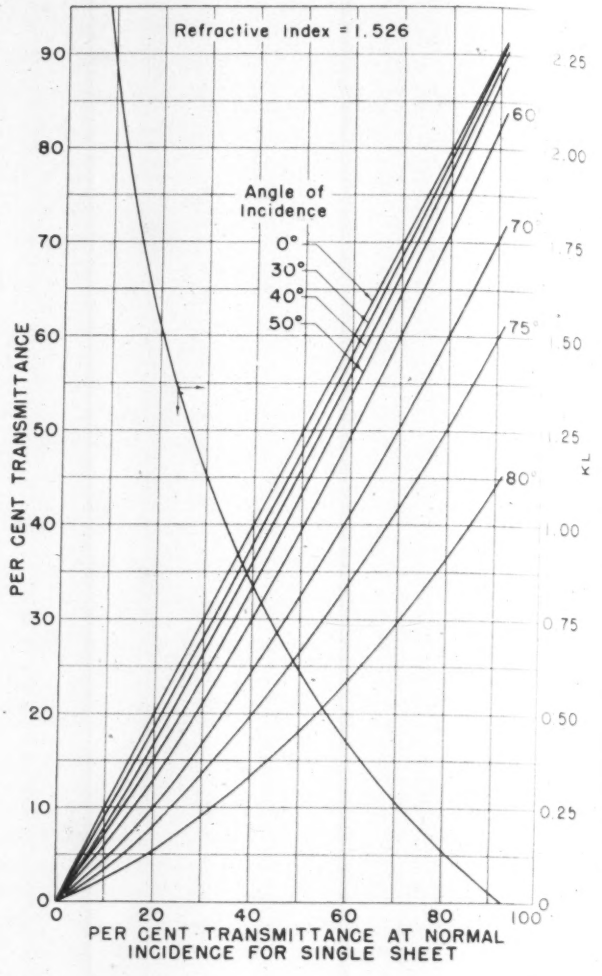

Fig. 9—Transmittance of single sheet of glass for direct solar radiation

The data of Fig. 8 have been similarly treated. On the date of test the sky was cloudless and considerably clearer than on September 18. However, because the solar altitude on the whole was lower (see Table 1), the solar intensities were also lower.

Calculated Transmittances of Single and Double Sheets

The Absorption Characteristic KL

It is of practical value to be able to predict the direct solar radiation transmittances of various glass combinations in a double glass installation from a knowledge of the transmittances of the sheets taken separately. It is also of interest to be able to predict the effects of glass thickness on transmittance of either single or double glass. The problem would be greatly simplified were reflectances the only thing to be considered. However, because many glasses absorb solar radiation to a greater or lesser extent and because the absorption coefficient varies with the wave length of the radiation, the problem presents some difficulties. To add to the difficulty the energy distribution in the solar spectrum varies with air mass, dust content, and water vapor content.

In an ASHVE Research Report curves were presented to show how the transmittance of single and double sheets of glass varied with the angle of the incident ray and the absorption characteristic KL, where K is the absorption coefficient in units per inch and L is the glass thickness in inches. These curves are presented in modified form as Figs. 9 and 10. The variable KL has been replaced by the transmittance for normally incident radiation. This relationship is expressed for single glass by Equation 6.

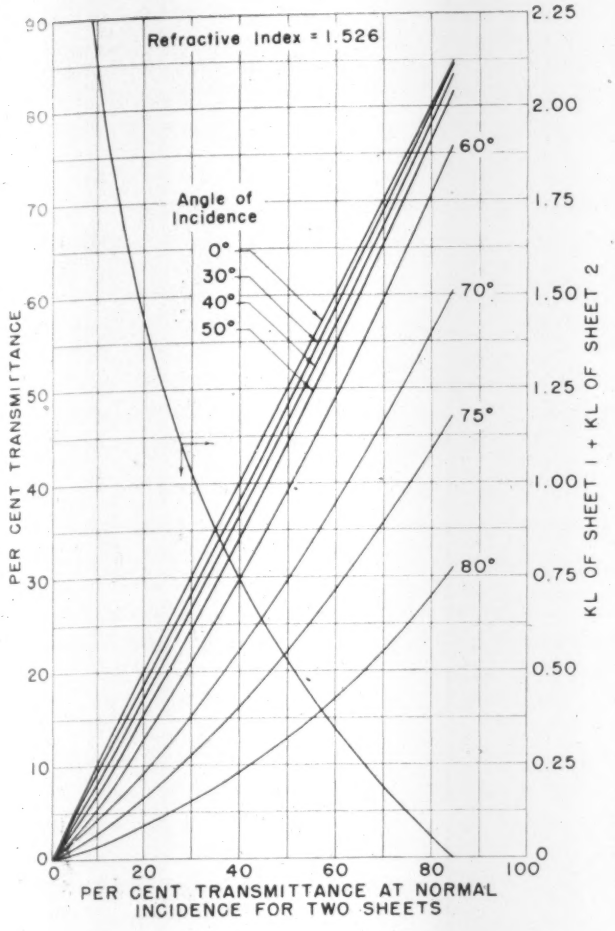

Fig. 10 — Transmittance of two sheets of glass for direct solar radiation

$$e^{-KL} = 1.094\ \tau_n = g' \quad \ldots\ldots (6)$$

where

e = base of natural logarithms

τ_n = transmittance at normal incidence

g' = fraction of the energy available after passage through the glass.

The index of refraction has been taken as 1.526. The value K is derived experimentally from the equation,

$$\frac{J}{J_o} = e^{-KL} \quad \ldots\ldots (7)$$

where

J = the intensity of the radiant beam which passes through one surface of a sheet of glass to the second surface,

J_o = intensity of the incident beam corrected for reflection at the first surface.

It should be noted that τ_n and J/J_o differ in that the former is the air-to-air transmittance and includes reflection losses at the glass-air interfaces. J/J_o is the surface-to-surface transmittance.

Transmittance Formulas

It has been shown[3] that for any incident angle the transmittance, τ_1, of single glass is expressed by the series,

$$\tau_1 = g\ (1-r)^2 + r^2g^3\ (1-r)^2 + r^4g^5\ (1-r)^2 + \ldots\ldots (8)$$

where

r = the Fresnel reflectance (note: this must be computed for each plane of polarization and depends upon the incident angle)

g = g' corrected for decreased path length due to refraction of the incident ray.

Summing the series,

$$\tau_1 = \frac{(1-r)^2 g}{1-r^2g^2} \quad \ldots\ldots (9)$$

As a result of summing a similar series, the transmittance, $\tau_{1,2}$, of two air spaced sheets of glass is expressed by the following equation:

$$\tau_{1,2} = \frac{\tau_1\tau_2}{1-\rho_1\rho_2} \quad \ldots\ldots (10)$$

where

τ_1 and τ_2 = transmittances of sheets 1 and 2

ρ_1 and ρ_2 = reflectances of sheets 1 and 2 computed by a method identical to that used to derive τ.

If $\rho_1\rho_2$ be neglected (at normal incidence $\rho_1\rho_2$ is of the order of 0.5 percent), then the transmittance of two sheets is approximately:

$$\tau_{1,2} = \tau_1\tau_2 \quad \ldots\ldots (11)$$

Comparison of Equation 11 with Equation 6 suggests that for two sheets of glass the sum of the KL values of the individual sheets can be used as a convenient means of determining transmittances of various combinations of glass in various thicknesses when the absorption coefficients are known, or:

$$\tau_{1,2} = \text{a function of } (K_1L_1 + K_2L_2) \quad \ldots\ldots (12)$$

If τ_n or J and J_o refer to the whole solar spectrum, then K (or KL) is an average value for the particular energy distribution associated with τ_n or J and J_o. On the other hand, if τ_n or J and J_o refer to a definite wave length (monochromatic radiation), then K has a fixed value for solar radiation of that wave length. It is apparent that Figs. 9 and 10 and Equations 9 and 10 can be exact only for monochromatic radiation.

Examples will now be given of the application of these curves. Assume that a sheet of glass ⅛ in. thick has, at a *certain wave length*, a normal incidence transmittance of 0.80. From Fig. 9, the KL value is 0.13. Its transmittances for radiation incident at 50 and 70 F are respectively 0.75 and 0.60. If the thickness were doubled the KL value would become 0.26 and the transmittance at normal incidence would become 0.71. The transmittances at 50 and 70 F would be 0.65 and 0.51.

Now assume that two air-spaced sheets of the same glass (⅛ in. thick) were to be used. The sum of the KL's of each sheet is 0.26. Fig. 10 shows that the corresponding normal incidence transmittance (for the same wave length as before) becomes 0.66. The transmittances at 50 and 70 F are 0.62 and 0.43. The procedure of determining transmittances of two sheets of glass by adding the KL values of each glass is a convenient one but

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it introduces a small error because $\rho_1\rho_2$, unlike $\tau_1\tau_2$, is not a function of $K_1L_1 + K_2L_2$. The error is not over 0.2 percent at 50 deg incident angle. At 75 deg transmittances determined by the use of Fig. 10 will be, at the most, 3 percent too low.

In a similar manner, Figs. 9 and 10 may be used to determine approximate transmittances for total solar spectrums. However, for most combinations of two sheets of glass, the normal incidence transmittance for the total spectrum cannot be determined with practical accuracy by adding individual KL values or by applying Equation 11. The errors involved and the magnitude of other deviations from the monochromatic curves are discussed in following sections. Certain approximations are suggested.

Discussion of Results

Effect of Energy Distribution at Normal Incidence

In a previous paper[3] calculations were made to determine the effect of energy distribution on the transmittance of single glass over the wave length band of the solar spectrum. It was found that solar energy distributed as suggested by Moon[2] for various air masses had no appreciable effect on the normal incidence transmittance of a glass such as No. 1 and that a maximum change of about 4 percent was produced in a glass similar to No. 10. No investigation was made with regard to double glass.

It is evident from the data given in Table 2 that solar energy distribution effects are in reality very significant and that no one value of normal incidence transmittance can be assigned to any given single glass or double glass combination. Exceptions to this are glasses such as No. 2 or No. 9 which have practically uniform transmitting characteristics over the entire spectrum except in the ultraviolet where the total energy is small. Furthermore energy distribution effects limit the application of Equations 10 and 11 (and Fig. 10). This can be illustrated by the following example.

Table 3—Transmittance at Normal Incidence for Two Sheets *vs* Computed Transmittance[a]

Date	Glass Combination	Observed Transmittance Percent	Calculated Transmittance, Percent $\tau_1 \times \tau_2$	Ratio Obs. / Calc.
3-25-48	Sgle #9	88.9		
	#9 In, #9 Out	79.1	79.0	1.00
3-24-48	Sgle #1	76.8		
	#1 In, #1 Out	60.8	59.1	1.03
10- 2-47	Sgle #1	75.2		
	#1 In, #1 Out	58.7	56.5	1.04
10-15-47	Sgle #1	75.5		
	Sgle #6	36.8		
	#1 In, #6 Out	32.8	27.8	1.18
10-23-47	Sgle #1	72.2		
	Sgle #10	35.6		
	#1 In, #10 Out	31.1	25.7	1.21
3-24-48	Sgle #6	41.3		
	#6 In, #6 Out	22.3	17.1	1.30
3-25-48	Sgle #10	36.3		
	#10 In, #10 Out	20.7	13.2	1.57

[a] Comparison of the normal incidence transmittances of combinations of two air-spaced sheets of glass with transmittances computed from the transmittances of the component sheets

Consider two identical sheets of glass having, for a given energy distribution, constant transmittances of 0.70 and 0.10 in the visible and infrared portions of the spectrum. Let it be further assumed that the energy contained in each of these parts of the spectrum is equal. The average transmittance of a single sheet for the whole spectrum is thus 0.40. Now if these two sheets are combined, Equation 11 would indicate that $\tau_{1,2}$ would be $\tau_1 \times \tau_2$ or 0.16. However, the first sheet would filter out so much infrared that the average transmittance of the second sheet for the beam now depleted of the infrared would be roughly 0.60. The combined transmittance would therefore be closer to 0.24 than 0.16. The following data (see Table 3) obtained by silver disk pyrheliometer tests illustrate this. They have been selected from tests which are included in Table 2. The transmittance of each combination and its components were determined within a period of 30 min—and, therefore, under approximately the same atmospheric conditions.

These data indicate the approximate error involved in computing transmittance by Equations 10 or 11 when τ_1 and τ_2 represent average transmittances for the whole solar spectrum. It will be noted that the greater differences are associated with those glasses which are least uniform in transmittance over the spectrum. It is evident that in order to completely describe the behavior of double glass combinations (1) many tests must be made to cover a wide range of conditions or (2) energy distribution curves must be set up and the transmittances computed wave length by wave length and an integrated result obtained.

It is, however, possible to predict approximately the transmittances of various combinations in one of two ways: (1) the transmittances for the whole spectrum can be computed by Equation 11 and a correction applied based upon data such as given in Table 3 or, (2) if the transmittances of each sheet in a given combination are given in terms of infrared and visible transmittance, the transmittance for each of these parts of the spectrum can be computed by Equation 11. If an energy distribution can be assumed, such as a ratio of infrared to visible radiation, the total normal incidence transmittance can be approximated. In either case, Fig. 10 can be used to obtain approximate transmittances at angles of incidence other than normal.

Energy Distribution Effects at Angles Other Than Normal Incidence

The series expression for transmittance of a single sheet (Equation 8) shows that the factor g, representing absorption, enters each term. For monochromatic radiation this expression is exact. However, over the spectrum as a whole, K, and hence g, may vary with wave length. With a glass such as No. 6

or No. 10, each successive reflected ray finds itself depleted of infrared so that the average K changes with each reflection. This effect is most significant in the infrared absorbing glasses and at the higher angles of incidence. The net effect is one of higher transmittance at these angles than the curves of Fig. 9 indicate. The same effect occurs in the case of double glass.

Examination of Equation 10 shows that the order in which the sheets are arranged should have no effect on transmittance. At any given wave length the products of τ_1 times τ_2 and of ρ_1 times ρ_2 are independent of order. The effect of energy distribution on transmission losses due to successive reflections does remain, however, and may result in significant differences due to order of arrangement. It should be noted that heat gain by convection and radiation from the inner glass *is dependent upon the order* in which the sheets are placed.

Comparison of Test Data with Theory

Because of the range of conditions under which the various glasses were tested (see Table 1) each solid curve of Fig. 6 must be considered as representing the average of a family of curves for that particular combination. The solid curves are curves of best fit. The dotted curves were constructed with the aid of Fig. 10. They are based upon monochromatic radiation and a KL value equal to that which represents the average absorption characteristic of a given combination for the total solar spectrum at normal incidence.

The actual deviation of a transmittance curve, calculated for an energy distribution corresponding to air mass = 2 from the monochromatic curve based upon the same normal incidence transmittance, was determined at two angles. For the double glass No. 1 combination the monochromatic curve was 0.5 percent and 3.5 percent below the true curve at 50 and 75 deg incident angles respectively. For the No. 1 and No. 6 combination, the deviations were 10 and 12 percent. Curves B and C are in reasonable agreement with this finding, although other energy distribution effects may account for this. Curve A deviates much more from the dotted curve than would be expected. In the case of double No. 9, the solid and dotted curves are identical.

Total Heat Gain

Figs. 7 and 8 illustrate total heat gain for two different double glass combinations. They also show rates of heat gain from the indoor surface of the inner sheet of glass. Also shown are data on the irradiation of the plane of the window. Fig. 8 demonstrates the decided reduction in heat gain when the outer glass is of the heat-absorbing type. Note that the heat transferred from the warmed inner glass in Fig. 8 is little different from that shown in Fig. 7. In both cases the elevation of the inner glass temperature above the calorimeter was about the same for roughly the same conditions, but in the case of combination No. 1 and No. 6, the temperature elevation of the outer glass above the outdoor air was two to three times greater than in the case of the double No. 1 combination. This caused a rapid dissipation of the absorbed solar heat. The test conditions were not, of course, strictly comparable because of the difference in wind velocity, which has considerable effect on heat dissipation at the outer glass surface. The total transmitted solar radiation is, of course, the total heat gain minus the heat flow from glass to calorimeter.

It should be noted that the curves of heat gain from glass to calorimeter lag with reference to the solar radiation curve by about one-half hour. This was found to be true of all similar fixed position tests of these and other combinations. Storage effects, therefore, are appreciable in glass installed in this manner. This storage effect, however, influences only the diffusion of solar heat absorbed by the glass and has no bearing on that part of the incident radiation directly transmitted through the glass.

Heat gain values represented by these curves must not be considered as design values. The calorimeter apparatus used in tests represents the extreme condition wherein cooling loads are immediately affected by transmitted solar radiation. In any practical installation interior walls, floors, and furnishings absorb much of the transmitted radiation and dissipate it to the room at reduced rates. Because the variables involved in treating such a problem are so numerous, this phase of the effects of solar heat gain must be considered separately.

The transference of heat from the inner surface of the inner glass to the calorimeter also represents a limiting condition, namely, one in which the interior surfaces seen by the glass and the ambient air are at substantially the same temperature. This condition is probably approached in many practical cases.

The solar energy transmittance data presented in Fig. 6 and Table 2 apply equally well to summer and winter conditions, subject, of course, to variations due to energy distribution effects. In winter the maximum heat gain would normally take place through southern windows rather than through western windows which is generally the case in the summer. The distribution of solar energy absorbed by the glass would probably differ from that shown in Figs. 7 and 8 because of differences in the temperature levels and wind velocity.

Summary

Transmittances of several types of glass used in various combinations of two sheets have been experimentally measured by calorimeter tests and by pyrheliometer tests. Total heat gains through these combinations with a ¼-in. air space have been measured under various conditions of temperature difference. The results of these tests are as follows:

1. Variations in solar energy distribution significantly affect the transmittance of these glasses for both direct and diffuse solar energy.

2. Because of these effects the transmittance of double glass is equal to or greater than the product of the transmittances of the component single sheets. For the combinations tested, this transmittance was about 5 percent greater for double standard plate glass and about 20 percent greater for stand-

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ard plate glass plus a heat-absorbing glass.

3. The use of a heat-absorbing glass outside of a regular plate glass reduces materially the directly transmitted solar radiation.

4. Under the conditions of the test setup the use of a heat-absorbing glass outside of a standard plate glass does not greatly increase the heat gain by radiation and convection from the inner glass as compared to the use of two sheets of standard plate glass.

The test results presented in this second report of studies on flat glass represent a part of a continuing program of research on windows and glass block panels.

Acknowledgment

The authors acknowledge with thanks the assistance and counsel given by their colleagues of the ASHVE Research Laboratory staff during the course of the work described in this report. In particular they wish to thank:

1. The American Window Glass Co., the Libbey-Owens-Ford Glass Co., the Mississippi Glass Co., and the Pittsburgh Plate Glass Co. for certain technical data.

2. The members of the Technical Advisory Committee on Glass for suggestions, advice, and technical information.

Bibliography

1. ASHVE Research Report, Measurements of Solar Heat Transmission Through Flat Glass, by F. V. Parmelee, W. W. Aubele, and R. G. Huebscher, (JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Vol. 20, No. 1, January 1948, pp. 158-166).

2. Proposed Standard Solar Radiation Curves for Engineering Use, by P. Moon, (Journal of the Franklin Institute, Vol. 230, No. 5, November 1940, pp. 583-617).

3. ASHVE Research Report No. 1281, The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by G. V. Parmelee, (ASHVE TRANSACTIONS, Vol. 51, 1945, pp. 317-350).

NEW CHAPTER AT SHREVEPORT

The Council has authorized the organization of a Chapter at Shreveport, La., and at a preliminary meeting on May 6, the Officers were elected and the appointment of standing committees was announced.

The Meetings Committee has announced that the Charter Meeting will be held on May 19 and presentation of the Charter will be made by a Member of the Council, Reg F. Taylor of Houston, and the principal speaker will be Carl F. Boester, Lafayette, Ind., on Reverse Cycle Refrigeration Combined with Radiant Heating as Applied to Small Homes.

Invitations to nearby Chapters were issued and it was anticipated that representatives from these groups would be in attendance.

The officers and committee chairmen will be announced in the July JOURNAL.

ENGINEERING EXPOSITION AT ST. PAUL

The Minnesota Chapter participated as an exhibitor in the Annual Engineering Exposition at St. Paul, Minn., in February 1948 and produced an exhibit telling about the aims and activities of the Society and of the Chapter. Arrangements for the booth were made by D. B. Anderson, and those who were in attendance included W. B. Buck, J. A. Craig, C. E. Gausman, H. E. Gerrish, L. J. Krause, O. L. Lilja, B. F. McLouth, J. F. Stafford, and William Sturm.

JOHN JAMES JOINS McDONNELL & MILLER

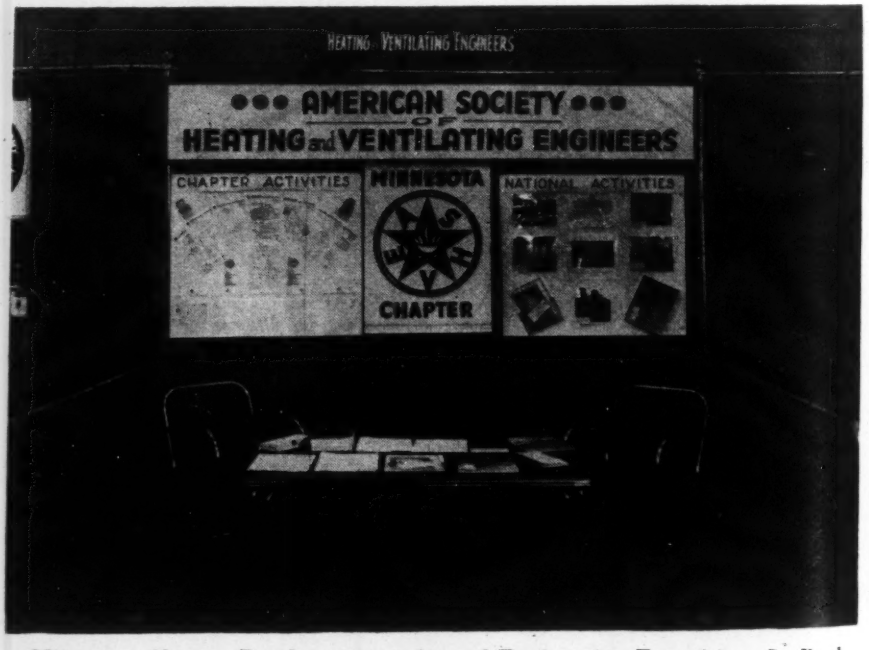
John James has joined McDonnell & Miller, Inc., Wrigley Bldg., Chicago, Ill., as chief research engineer.

Mr. James was born at Brookville, Ind. He attended the Benson Polytechnic School, Portland, Ore., was graduated from Oregon State College with a B.S. and received his Master's degree in mechanical engineering from the University of Wisconsin in 1934.

Specializing in heating, ventilating and air conditioning, Mr. James has served as engineer for the General Electric Co., Schenectady, N. Y., and in 1935 joined the staff of the ASHVE as technical secretary, resigning in 1943 to join Iron Fireman Mfg. Co., Cleveland, Ohio.

During his years with the Society his responsibilities included the preparation of the technical data for THE GUIDE, the correlation and administration of the cooperative research program, as well as work with the technical and code committees. As a member of the ASHVE he is serving as chairman of the Publication Committee, as a member of the Committee on Constitution and By-Laws, and on the Committee on Research. He is now completing his term as president of the Northern Ohio Chapter.

Mr. James is co-author with John R. Allen and J. Herbert Walker of the book *Heating and Air Conditioning*.



Minnesota Chapter Booth at recent Annual Engineering Exposition, St. Paul

Veranda of Mount Washington Hotel

Bretton Woods Welcomes ASHVE

FINAL arrangements have been made by the committee of the Massachusetts Chapter in charge of events at the Bretton Woods meeting of the Society to be held June 20-23 at the Mount Washington Hotel. A meeting of the Council and several committee meetings will be held on Sunday, June 20. There will be three technical sessions arranged by the Program Committee, of which A. E. Stacey, Jr. is chairman.

At the first session on Monday, June 21, two important matters of business will be presented to the members in the form of amendments to the By-Laws changing the dues rate and revision of Regulations Governing the Committee on Research.

Technical sessions will have papers on physiological factors in air conditioning, heat storage problems, and solar heat transmission. There will be nine papers and an interesting talk on Mount Washington weather conditions by Prof. Donald Chapman.

The Nominating Committee will meet to select candidates for office in the Society and the Chapter Delegates will hold two meetings—one on Monday afternoon and the other on Tuesday evening.

C. M. F. Peterson, general chairman of arrangements, has announced a special program for the ladies and some interesting sightseeing tours. There will be a cog rail trip up Mount Washington on Monday afternoon if weather permits and, as an alternate, a trip to Franconia Notch with a visit to the Old Man of the Mountains, Profile Lake, and the Flume, with a ride on the aerial tramway to Cannon Mountain.

On Tuesday for those who do not participate in the golf tournament there will be a cog rail trip or a trip to Crawford Notch with a ride to Cranmore Mountain on the skimobile.

For those who want entertainment on Wednesday afternoon there will be a soft-ball game, a golf tournament, or a trip to Pinkham Notch.

For evening entertainment, on Monday a barn dance is planned, on Tuesday special games and dancing, and on Wednesday evening the banquet will be held with Prof. James Holt, Boston, as toastmaster and some of the world famous New Hampshire stories will be related by Laurence Whittemore, president of the Federal Reserve Bank of Boston.

For members traveling by train who desire to arrive in Bretton Woods on Sunday the Massachusetts Chapter Committee on Arrangements is planning to have a chartered bus leave Boston at 11:30 a.m., arriving at Bretton Woods at 5:30 p.m. This will provide a scenic route for those coming to Boston on early morning trains on Sunday, June 20.

Railroad tickets may be obtained via Springfield, Boston, or New York. Local ticket agents will be glad to quote fares and rates.

The chartered bus will pick up passengers at the South Station and at the Statler Hotel. Advance reservations for this trip are required and the round trip fare will be $10.00 per person.

For the convenience of members who wish to arrive at Bretton Woods on Saturday or Sunday, June 19 and 20, the accompanying typical schedules are given.

Those who wish to drive will find that the following routes will generally provide good road conditions and a scenic trip through the New England states or via the Adirondacks:

Motor Routes to Bretton Woods, N. H.

BY CAR

1. *From the West to Troy, N. Y.:*
   - Route 2—Mohawk Trail to Greenfield, Mass.
   - Route 5—through Wells River, Vt., cross Connecticut River to Woodsville, N. H.
   - Route 302—to Bretton Woods
2. *Alternate Route:*
   - Route 9N—to Au Sable Forks, Ferry from Ft. Kent to Burlington, Vt.

Typical Railroad Schedules

For arrival in Bretton Woods, N. H. June 19 or 20 *only*

			(Via Boston)			
1.	*Lv.* Chicago	June	18	2:30 p.m.	(C.S.T.)	N. Y. Central
	ar. Boston	"	19	9:55 a.m.	(E.S.T.)	
	lv. Boston	"	19	11:00 a.m.	(E.S.T.)	Boston & Maine R.R.
	ar. Bretton Woods	"	19	3:30 p.m.	(E.S.T.)	
2.	*lv.* Chicago	June	19	2:30 p.m.	(C.S.T.)	N. Y. Central—Train 28
	ar. Boston	"	20	9:55 a.m.	(E.S.T.)	
	lv. Boston	"	20	11:30 a.m.	on charter	B. & M. Bus
	ar. Bretton Woods	"	20	5:30 p.m.	(C.S.T.)	
3.	*lv.* St. Louis	June	19	9:10 a.m.	(C.S.T.)	N. Y. Central—Train 12
	lv. Indianapolis	"	19	1:45 p.m.	(C.S.T.)	
	lv. Cleveland	"	19	8:20 p.m.	(E.S.T.)	
	lv. Buffalo	"	19	11:55 p.m.	(E.S.T.)	
	ar. Boston	"	20	9:55 a.m.	(E.S.T.)	
			(Via New York)			
4.	*lv.* Detroit	June	19	8:20 a.m.	(E.S.T.)	N. Y. Central
	lv. Cleveland	"	19	9:50 a.m.	(E.S.T.)	
	lv. Buffalo	"	19	1:30 p.m.	(E.S.T.)	
	ar. New York	"	19	9:45 p.m.	(E.S.T.)	
	lv. New York	"	19	11:10 p.m.	(E.S.T.)	N. Y. N. H.-H. R.R.
	ar. Whitefield	"	20	1:50 p.m.	(E.S.T.)	B. & M. R.R.
5.	*lv.* Chicago	"	18	1:30 p.m.	(C.S.T.)	N. Y. Central
	lv. Cleveland	"	18	8:00 p.m.	(E.S.T.)	
	lv. Buffalo	"	18	10:55 p.m.	(E.S.T.)	
	ar. New York	"	19	7:20 a.m.	(E.S.T.)	
	lv. New York	"	19	8:10 a.m.	(E.S.T.)	N. Y. N. H.-H. R.R.
	ar. Whitefield	"	19	5:55 p.m.	(E.S.T.)	B. & M. R.R.

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Route 2 —to Montpelier, Vt.
Route 302—to Wells River, Vt., cross river to Woodsville, N.H. and continue on 302 to Bretton Woods

3. *From South and West via New York City:*
Hutchinson River Parkway & Merritt Parkway to New Haven, Conn.
Route 5 —to Hartford
Route 5A—to Springfield
Route 5 —to Brattleboro and Wells River, cross Connecticut River to Woodsville, N. H.
Route 302 —to Bretton Woods

4. *From Boston:*
Route 2—to Junction of Route 4 at Lexington
Route 4—to Nashua and Twin Mountain
Route 302—to Bretton Woods

5. *Alternate Route*
Route 1—to Portsmouth, N. H.
Route 16—from Portsmouth to Dover, Rochester & Glen
Route 302—to Bretton Woods

Hotel reservations should be sent to the Mount Washington Hotel, Bretton Woods, N. H. and r[illegible]es on the American Plan for the m[illegible]eting are $13.00 per person per d[illegible]y fo. double and twin bed room[illegible] with bath; $15.00 per person per [illegible]y for- single room with bath; and $12.00 per person per day for si[illegible]le or double room with running w[illegible]ter.

The officers and memb[illegible]s of Massachusetts Chapter loo[illegible] forward to greeting all memb[illegible]rs at Bretton Woods in the heart [illegible]f the White Mountains, June 20-2[illegible]

Committee on Arrangements

C. M. F. Peterson, *General Chairman*
James Holt, *Vice Chairman*
Banquet—Hugo Von Rehberg
Entertainment—C. W. Larson
Finance—J. W. Brinton
Ladies—Adolph Ehrenzeller
Publicity—D. W. Blair
Reception—E. A. Dusossoit
Sports—A. C. Bartlett
Transportation—W. H. Shipp

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Reception

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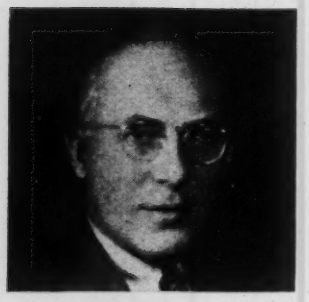
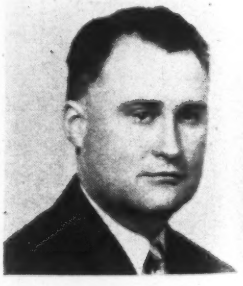
Banquet



A. C. Bartlett
Sports

J. W. Brinton
Finance



W. H. Shipp
Transportation

PROGRAM

Semi-Annual Meeting 1948

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Mount Washington Hotel
Bretton Woods, N. H.

June 20-23, 1948

Sunday—June 20

10:00 a.m. Committee on Research Meeting

1:00 p.m. Registration

1:30 p.m. Council Meeting

Monday—June 21

9:00 a.m. Registration

10:00 a.m. TECHNICAL SESSION—Physiological Factors in Air Conditioning
Greetings by C. M. F. Peterson, General Chairman
Response by Pres. G. L. Tuve
Report of Committee on Research—L. N. Hunter, Chairman
Amendments
1. By-Laws
2. Regulations Governing Committee on Research
Sanitary Significance of Ventilation, by W. F. Wells
The Mount Washington Observatory—America's Original Mountain Weather Station, by Prof. Donald H. Chapman
Thermal Insulation of Clothing, by C. P. Yaglou
Comparison of Physiological Adjustments of Human Beings During Summer and Winter, by Nathaniel Glickman, Tohru Inouye, Robert W. Keeton and Maurice K. Fahnestock

10:30 a.m. Croquet and Quoits Tournament and Golf for Ladies

12:30 p.m. Get-together Luncheon

1:30 p.m. Guide Publication Committee Meeting

1:30 p.m. Chapter Delegates Meeting

1:30 p.m. Cograil trip to Mount Washington (Bus to Base Station)

or

Trip to Franconia Notch, visiting the Old Man of the Mountains, Profile Lake and the Flume, with ride on Aerial Tramway to Cannon Mountain
Informal Golf—Tennis

6:30 p.m. Dinner

8:00 p.m. Nominating Committee Meeting

8:30 p.m. Movies—Barn Dance

Tuesday—June 22

9:00 a.m. Registration

9:30 a.m. TECHNICAL SESSION—Heat Storage
Ground Temperature Distribution with a Floor Panel Heating System, by A. B. Algren
Theory of the Ground Pipe Heat Source for the Heat Pump, by L. R. Ingersoll and H. J. Plass
Fan Selection by Use of Constants, by A. C. Stern and Leon D. Horowitz

On the Links of Mount Washington Hotel

10:00 a.m. Ladies Golf—Quoits, Croquet—Hike up Mt. Willard

1:30 p.m. Golf Tournament for Research and Eichberg Memorial Cups
Fishing trip

1:30 p.m. Sightseeing Automobile Trip to Crawford Notch, visiting Willey House, North Conway and trip up Cranmore Mountain on Ski-Mobile

6:30 p.m. Dinner

8:00 p.m. Chapter Delegates Meeting

8:30 p.m. Games and Dancing

Wednesday—June 23

9:00 a.m. Registration

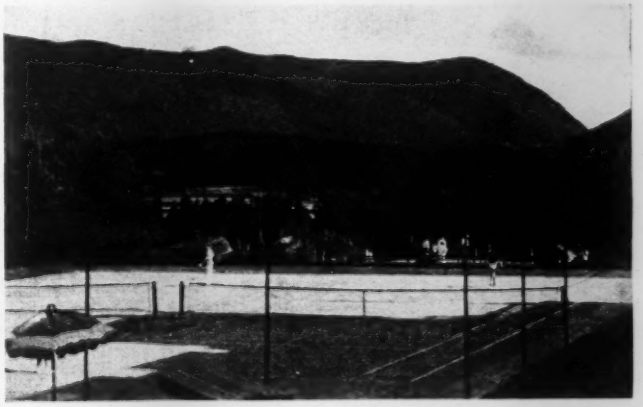
9:30 a.m. TECHNICAL SESSION—Solar Heat Gain Through Walls and Roofs for Cooling Load Calculations, by J. P. Stewart
Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation, Convection and Conduction, by C. S. Leopold
Solar and Total Heat Gain through Double Flat Glass, by G. V. Parmelee and W. W. Aubele

10:30 a.m. Ladies Card Party—Golf or Croquet

12:30 p.m. Luncheon

2:00 p.m. Soft Ball, Baseball Game, Golf, Swimming, Tennis
Motor Trip to Pinkham Notch Circle, Presidential Mountain Range

7:00 p.m. Banquet
Toastmaster—Prof. James Holt
Awarding of Golf Trophies
Address by Laurence Whittemore, President of Federal Reserve Bank of Boston



Tennis Courts of Mount Washington Hotel

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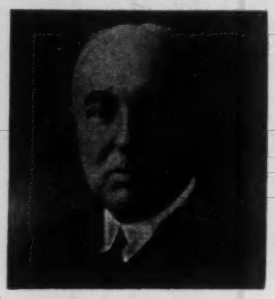


C.-E. A. Winslow,
Past President, 1945
New Haven, Conn.

C.-E. A. WINSLOW, president of the Society in 1945, is the editor of the American Public Health Association Journal. For many years he was associated with Yale University, School of Medicine, New Haven, Conn. Dr. Winslow was formerly director of the John B. Pierce Laboratory of Hygiene. He has served the Society in many capacities and is also well known as the author of scientific papers and books on health and comfort.

E. W. Klein
Atlanta, Ga.

EDWARD W. KLEIN*, sales engineer, Atlanta, Ga., has been active in the design and sales of heating and ventilating equipment for nearly 30 years. He graduated from the Georgia School of Technology in 1902 with a BSME degree. For many years he has been a member of the ASHVE and *ASME* and has served as president of the local chapters of both societies.

*Deceased April 21, 1948

E. C. Evans
Temple City, Calif.

EDWIN C. EVANS, Temple City, Calif., retired from his position as manager of the B. F. Sturtevant Co. Buffalo office in 1945, after 10 years' service with the company. For many years previous to 1936 Mr. Evans was with American Blower Corp. He has been active in the formation of the Southern California and St. Louis Chapters of the ASHVE, and served as president of the Pittsburgh Chapter. Mr. Evans is a graduate of Kentucky State.

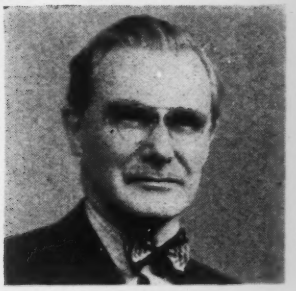
CHARLES L. DEAN, chairman of the mechanical engineering department, extension division, University of Wisconsin, Madison, Wis., has been connected with the University for 22 years. Since graduating in 1901 with a BS in mechanical engineering from the University of Wisconsin, he has specialized in heating, ventilating, and refrigeration work.

C. L. Dean
Madison, Wis.

THOMAS F. McCOY has been associated with the Powers Regulator Co., Boston, Mass., for 43 years. In 1907 he was appointed manager of the New England territory, a position which he holds at the present time. During this time he has supervised numerous automatic temperature control installations in private residences, churches, schools, colleges, hospitals, theaters, and many industrial plants.

T. F. McCoy
Boston, Mass.

Election to membership 1948



S. E. Dibble
Past President, 1925
Elizabethtown, Pa.

SAMUEL EDWARD DIBBLE, president of the ASHVE during 1925, is superintendent of the Thomas Ranken Patton Masonic Institution for Boys, Elizabethtown, Pa. Prior to his present position he served as a professor of Heating, Ventilating and Plumbing at the Carnegie Institute of Technology. An editor and writer, Mr. Dibble has contributed to engineering publications and is the author of *Elements of Plumbing* and the *Plumbers' Handbook*.

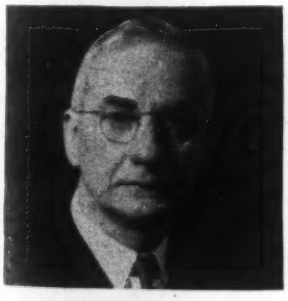
H. E. Gerrish
Minneapolis, Minn.

HARRY E. GERRISH has served as president of the Morgan-Gerrish Co., Minneapolis, Minn., manufacturers' representatives since 1911. In 1944 Mr. Gerrish was appointed regional engineer of the national fuel efficiency program. Mr. Gerrish has been active in the affairs of the ASHVE and helped organize the Minnesota Chapter of the Society. He holds a mechanical engineering degree from the University of Minnesota.

Arthur Ritter
New York, N. Y.

ARTHUR RITTER is the New York district manager of the American Blower Corp. He has been associated with this company since 1902, serving in the various capacities of draftsman, salesman, branch manager, district manager, manager export department, and officer and director. A charter member of the ASHVE New York Chapter, Mr. Ritter has served as its president, vice president and treasurer.

JOHN C. WHITE, steam and mechanical engineer, Madison, Wis., retired in 1944 from his position as state power plant engineer, Wisconsin State Bureau of Engineering, a position which he had held since 1915. Among his many achievements in the engineering field is the design of a jet system of heating known as the *Hydroheater*. He was awarded an honorary mechanical engineering degree by the University of Wisconsin in 1928.



R. R. Taliaferro
Cleveland, Ohio

ROBERT R. TALIAFERRO is a mechanical engineer for the National Advisory Committee for Aeronautics, Aircraft Engine Research Laboratory, Cleveland Airport, Cleveland, Ohio, where he is active in the operation of a refrigeration and air conditioning plant. He was formerly associated with the Carrier Corp. for a number of years. In 1913 he received his mechanical engineering degree from the University of Kentucky.

J. C. White
Madison, Wis.

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J. F. McIntire
Past President 1939
Detroit, Mich.

JAMES F. McINTIRE, Detroit, Mich., formerly president of U. S. Radiator Corp., Detroit, Mich., is now retired. He joined the U. S. Radiator Corp. staff in 1910 as a boiler engineer, after graduating from the University of Illinois in 1907 with a B.S. degree. Active in the interests of the Society, Mr. McIntire served in many important positions including that of president in 1939, and on the Council 1926-28 and 1932 to 1940.

THOMAS CHESTER, consulting engineer, Birmingham, Mich., was associated with a number of firms in the manufacturing, contracting, and consulting fields until his retirement in 1944. During his many years in the engineering profession Mr. Chester was granted 18 patents in connection with air engineering and air conditioning. He has contributed many papers to the various engineering publications and in 1944 was awarded the Silver Medal of the *Institution of Heating and Ventilating Engineers* of Great Britain.

HENRY NOTTBERG, SR., president of the U. S. Engineering Co., Kansas City, Mo., previously was vice president and secretary-treasurer of the company. A member of the U. S. Engineering Co. staff since 1905, he has been active in designing and installing heating, ventilating and refrigeration equipment. Mr. Nottberg joined the ASHVE in 1919 and served as a member of the Board of Governors of the Kansas City Chapter.

ARTHUR S. LEITCH is president and managing director of the Arthur S. Leitch Co., Ltd., Toronto, Ont., Canada. Mr. Leitch has been active in the organization of the Ontario Chapter, serving as its president in 1924. For over 50 years he has been prominent in the heating and ventilating profession in Canada.

JAMES POSEY, Baltimore, Md., established his consulting firm in 1911. Since that time he has been responsible for the installation of heating, ventilating, electric wiring and other mechanical equipment for many important buildings in the Baltimore area. Mr. Posey received his engineering training at John Hopkins University. Mr. Posey became a member of the Society in 1919.

E. VAN WAGENEN NOW CONSULTING ENGINEER

Edward Van Wagenen has announced the opening of his consulting mechanical engineering office located at 119 Newnan St., Jacksonville, Fla.

Prior to the opening of his new office Mr. Van Wagenen was a mechanical engineer at the U. S. Naval Air Station in Jacksonville. He is a graduate of the University of Virginia and is a member of the ASHVE.

F. E. INCE JOINS MARLO COIL CO.

F. Edward Ince, St. Louis, Mo., has joined the Marlo Coil Co., as sales application and research engineer. Mr. Ince was formerly with the Kennard Corp., St. Louis, as vice president and has had extensive engineering experience in the cooling field.

Mr. Ince graduated from the University of Missouri in 1931 with an electrical engineering degree and had previously taken an electrical engineering course at Cooke Engineering School. He joined the ASHVE in 1944.

J. C. BARNES ELECTED VICE PRESIDENT

The election of John C. Barnes as vice president for sales, The National Radiator Co., Johnstown, Pa., has been announced by Robert S. Water, president of the company. Mr. Barnes, who has been associated with the firm since 1940, was formerly sales manager, first of the company's Washington branch and later of the Philadelphia branch office.

For seven years prior to joining the National Radiator Co., Mr. Barnes was associated with Montgomery Ward and Co., Inc. He was elected an associate member of the ASHVE in 1944.

J. C. Barnes
Johnstown, Pa.

OGDEN & WOODRUFF CONSULTING ENGINEERS

The consulting engineering office of Ogden & Woodruff, 3133 North Boulevard, Baton Rouge, La., has been formed by H. N. Ogden and C. S. Woodruff.

Prior to the formation of his own office, Mr. Woodruff was associated as an engineer with the firm of Reg. F. Taylor, consulting engineer. He has been active in design and application engineering since his graduation from Northwestern Technical Institute in 1935. Mr. Woodruff holds membership in the ASHVE and his partner, Mr. Ogden, is a member of the *ASCE*.

COUNCIL MEETING AT THE LABORATORY

The April 19 meeting of the Council was held at the Research Laboratory, Cleveland, Ohio, and all but three members were in attendance as shown in the accompanying picture. It was announced that most of the available space for the Exposition at Chicago in January, 1949, has been sold. An invitation from the Minnesota Chapter to act as host for the Semi-Annual Meeting in June 1949 was accepted. Favorable consideration was given to the North Texas Chapter's invitation to hold the Annual Meeting in Dallas in January 1950, and investigation is now being made regarding Exposition facilities and hotel accommodations.

The Membership Committee presented a favorable report and indicated that the new Shreveport Chapter would hold its charter meeting shortly. The Council granted charters for Student Branches at North Carolina State College, Raleigh, N. C., and at Purdue University, W. Lafayette, Ind.

The Finance Committee presented an extensive report and a revised budget for 1948. Some amendments were proposed for revising the rules governing the Committee on Research as they relate to finances, and there was considerable discussion about a return to the former dues rate of $25.00 for members and associates of the Society in 1949.

It was proposed that a digest of the Council Minutes be sent to all members of the Society with the President's letter.

C. G. BINDER RETIRES

Announcement has been made of the retirement of Charles G. Binder, chief engineer of application at

C. G. Binder
Merchantville, N. J.

Warren Webster & Co., Camden, N. J., since 1928. Associates of Mr. Binder honored him at a farewell dinner, and presentation of a set of golf clubs was made by C. J. Kern, his assistant, who succeeds him as manager of application engineering and service.

Early in May, Mr. and Mrs. Binder sailed for an extended visit to European countries including France, Belgium, Holland, Switzerland and the American zone in Germany. The Binders, who have devoted much of their free time since the war in helping needy families abroad, recently mailed their 465th box of food and drugs.

POSTHUMOUS AWARD FOR J. HERBERT WALKER

The late J. Herbert Walker was among the five Detroit Edison Co. employees who were awarded an Alex Dow Medal, in recognition of an unusual achievement or outstanding contribution related to company operations and responsibilities during 1947. Mr. Walker was posthumously honored for his outstanding contribution to the field of human relations as typified by *The Question Box*, a monthly feature of the company's magazine *Synchroscope*, which he originated and developed to answer employee questions. At the time of his death on December 1, 1947, Mr. Walker was vice president of the Detroit Edison Co., and had served the utility for 36 years.

Other Detroit Edison employees honored with an Alex Dow Medal are Marvin Horn, Robert Aiken, Thomas Daly and J. R. Hamann.

Council Meeting in Cleveland

Seated L. to R.: **D. M. Allen, C. Rollins Gardner, Carl F. Boester, F. W. Hutchinson, R. A. Sherman, M. W. Bishop, L. T. Avery, L. E. Seeley, G. L. Tuve, A. V. Hutchinson, C. S. Leopold, Miss Dorothy Mildner, Earl G. Carrier, M. S. Wunderlich, F. A. Hamlet, H. E. Sproull, R. F. Taylor, L. N. Hunter.**
Standing: **C. A. McKeeman, Cyril Tasker**

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OREGON CHAPTER CONFERENCE

The fifth Oregon State Air Conditioning Conference sponsored by the Oregon Chapter of ASHVE was held at Benson Polytechnic School, in Portland, March 31-April 3 inclusive, and included meetings that were open to the public. All previous attendance records were broken, with an attendance of 1650 at the 12 technical meetings. Registrations for technical sessions totaled 377, of which 139 were Portlanders, not chapter members; 99 were from Oregon, outside the Portland metropolitan area; 53 were from Washington (state), 22 of whom came from Seattle. Attendance at the open sessions and at the public Exposition could only be approximated.

The technical papers, carefully prepared for the occasion, were outstanding in their treatment of problems of current interest and importance. In general, they were designed to furnish important practical information for application in our profession. Discussions resulted in bringing out the full usefulness of the papers and in focusing attention on wider fields.

C. W. Brissenden, president, Oregon Chapter, receives bronze book ends

The Chapter was fortunate in having a Member of Council of the Society at the Conference, as a speaker, Prof. F. W. Hutchinson, Professor of Mechanical Engineering, University of California.

The display of equipment was the first such public display of heating equipment in Portland; it was crowded with eager spectators, especially in the evenings. Many paid a second visit. The neat, artistic exhibits were arranged against attractive backdrops that accentuated the important pictures.

A not unimportant feature of the Conference was the more informal get-togethers where members of the profession became better acquainted in discussing their merchandising and technical problems, and enjoyed the friendly fellowship for which the occasion offered a rare opportunity. There was also a sparkling banquet, which was an ideal occasion for the promotion of real friendship and fraternization. In fact, this was one of the most valuable features.

The Chapter owes a large debt to all the following: W. A. Simpson, chairman of the Conference Committee, and to all who assisted him; T. H. McClung, chairman, and his Attendance Committee; K. H. Hanson, chairman, and his Publicity Committee; W. Bruce Morrison, chairman, and his Technical Program Committee; W. H. Stover, chairman, and his Banquet and Annual Meeting Committee; J. H. Bonebrake, and his Committee on Arrangements; and T. E. Taylor, chairman, and his Special Committee on Ways and Means, without whom the other groups could not have functioned at all.

Oregon Chapter members and guests attending conference at Benson Polytechnic School, Portland, Ore.

Summary of Local Chapter Meetings *

ATLANTA—April 9, 1948. Subject: Axial Flow Fans. Speaker: W. R. Heath, chief engineer, Buffalo Forge Co., Buffalo, N. Y. Mr. Heath's very interesting presentation was illustrated with slides. Other Features: Reports were given by G. B. Hightower, chairman of the attendance committee, and E. K. Jamison, chapter treasurer. Pres. L. L. Barnes announced that the society's Life Membership Certificate had been presented to Edward W. Klein and a motion was carried that Mr. Klein also be awarded life membership in the Atlanta Chapter. Attendance 48. Attendance ratio 0.26.

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CENTRAL OHIO—April 10, 1948. Subject: The ASHVE Research Laboratory. Speakers: Cyril Tasker, director of research, and G. V. Parmelee, research fellow, ASHVE Research Laboratory. Mr. Tasker, in a talk entitled, The Challenge of the ASHVE Research Program, discussed projects which are under way at the society's Research Laboratory. Following Mr. Tasker, Mr. Parmelee discussed, in detail, research on solar radiation transmission. Tests on this subject are now under way at the Laboratory. Both talks evoked a great deal of interesting comment and were enjoyed by all. Attendance 33. Attendance ratio 0.32.

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CENTRAL NEW YORK—April 21, 1948. Subject: Test Results Conducted in Two Test Residences at the University of Illinois. Speaker: Prof. W. S. Harris, University of Illinois, Urbana, Ill. One of the test houses constructed at the University is a fully insulated two-story, six-room house with a full basement. The second building is a prefabricated single story, five-room house, built on a concrete slab floor laid directly on the ground. The data presented by Professor Harris showed comparative performance of the two types of radiant baseboard panels and small tube conventional type radiators. Other Features: Reports were given by the treasurer, the chapter delegate to the Annual Meeting and the chairman of the membership committee. Officers for the coming year were elected as follows: D. L. Mills, *president;* V. S. Day, *vice president;* A. L. Jones, *treasurer;* J. H. Carpenter, *secretary;* and H. G. Strong, L. E. LaRow and R. E. Cherne, members of the *board of governors.* Attendance 51.

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CINCINNATI—March 2, 1948. Subject: Activities of the Research Laboratory. Speakers: Cyril Tasker, director of research, and G. V. Parmelee, research fellow, ASHVE Research Laboratory. Mr. Tasker gave a very interesting report on what the Laboratory is doing and what is planned for the future. After his talk, Mr. Parmelee spoke on solar radiation transmission through glass, one of the subjects which are now being studied at the Research Laboratory. Other Features: Reports were given by the chairmen of the membership and entertainment committees. H. K. Jennings, chapter president and delegate to the 54th Annual Meeting, reported briefly on events at the Meeting and stated that a detailed written report would be given to each member so that he could digest it at his convenience. Attendance 43.

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CONNECTICUT—March 11, 1948. Subject: Developments in the Art and Industry of Air Conditioning. Speaker: A. E. Stacey, Jr., first vice president of the society and process consultant, sales division, Carrier Corp., Syracuse, N. Y. Pres. S. R. Osborne thanked Mr. Stacey on behalf of those present. Other Features: After an announcement was made that the wife of H. C. Kelly had passed away, the secretary was instructed to extend the sympathy of the chapter to Mr. Kelly. A letter regarding emblems for chapter past presidents was read and a motion carried that the treasurer order a sufficient number of the emblems to be presented to each past president. The 64th annual meeting of the *Connecticut Society of Civil Enginers* scheduled for March 16 and 17 was announced. Attendance 50. Attendance ratio 0.3.

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DELTA—March 9, 1948. Subject: Use of Germicidal Lamps in Air Conditioning Spaces. Speaker: G. E. May, principal utility engineer, New Orleans Public Service, Inc., New Orleans, La. Mr. May's talk was interesting and informative and was thoroughly enjoyed by all. Other Features: A film entitled *Operations Crossroads,* dealing with the Navy's atomic bomb tests which were made both over and under water at Bikini Atoll, was shown. Attendance 57.

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GOLDEN GATE—April 1, 1948. Subject: Ladies Night. Two Films were shown, a Standard Oil Film entitled, *San Francisco, Queen of the West,* and a Minneapolis Honeywell production entitled *Basic Electronics.* Other Features: John Everetts, Jr., chapter delegate to the 54th Annual Meeting reported on recommendations made at the Meeting. Mr. Everetts also discussed the financing of the Society's Research Laboratory and the exhibits at the Exposition. The nominating committee presented its recommendations for officers for the coming year. Pres. F. W. Kolb announced a committee had been appointed to cooperate with the Southern California Chapter in connection with the Heat Transfer and Fluid Mechanics Institute. Members of the committee

*Note: The attendance ratios shown represent the membership attendance divided by the chapter membership. These ratios will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by the various chapters and may be useful in deciding on subjects for chapter meetings.

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were named as follows: Edward Simons and O. W. Johnson; with F. L. Nemec and K. O. Alexander serving as alternates. Attendance 91.

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GOLDEN GATE—March 4, 1948. Subject: Air Distribution, Its Problems and Solutions. Speaker: J. B. Hewett, sales manager, Anemostat Corp. of America, New York, N. Y. Mr. Hewett stated that probably 50 percent of the unsatisfactory ventilating jobs are caused by problems of air distribution. Slides and moving pictures of laboratory tests and illustrations of air diffusers were shown. Mr. Hewett concluded with some graphic displays on a working model, using smoke tests. Other Features: A nominating committee was appointed and requested to report at the next meeting. A discussion was held on securing additional material for the chapter's publication, *The Fog Dispenser*, and reporters were named to collect new items from the various local sections. A communication from the Oregon Chapter announcing their Annual Conference scheduled in Portland March 31, April 1, 2 and 3, was read. Attendance 89.

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ILLINOIS—April 12, 1948. Subject: Annual Social Meeting. The 22 past presidents who were present as guests of the chapter were introduced by Pres. W. A. Kuechenberg. The meeting was devoted to entertainment features which included sound movies of the 1947 world series and all star baseball games. Vincent Gottschalk, magician and mind reader, also entertained the guests, and the members and guests thoroughly enjoyed the performance. Attendance 117.

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INDIANA—March 26, 1948. Subject: Heating Methods. Speakers: W. W. Grear, engr., Freyn Bros., Inc.; W. R. Shuttleworth, Shuttleworth Conditioned Air Co.; and S. E. Fenstermaker, Jr., partner, S. E. Fenstermaker & Co., Indianapolis, Ind. Mr. Grear gave a short discussion on the heat pump. He was followed by Mr. Shuttleworth who gave a talk on his practical experience with various types of radiant heating installations. The last speaker was Mr. Fenstermaker, who discussed baseboard radiation. This type of program proved to be most interesting and highly successful. Other Features: G. B. Supple gave a report on the by-laws committee, and P. R. Jordan reported on the research fund. Pres. I. W. Cotton appointed members to serve on the nominating committee. A short talk on the proposed Purdue Student Branch of the ASHVE was given by Professor Morris of Purdue University. C. M. Burnam, Jr., editor, *Heating, Piping and Air Conditioning*, gave a brief explanation of the coordination between the Society and the publisher. Attendance 87.

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INDIANA—February 27, 1948. Subject: The History of Air Conditioning. Speaker: W. A. Danielson, president, Danielson Associates Inc., Memphis, Tenn. A discussion period followed Mr. Danielson's talk. Other Features: T. R. Davis gave a complete report on the chapter delegates' meeting held at the 54th Annual Meeting of the Society. Pres. I. W. Cotton again urged chapter members to contribute toward the fund to reduce the mortgage on the Society's Research Laboratory building. President Cotton also suggested that the chapter consider organizing a student branch at Purdue University. Attendance 40.

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IOWA—April 13, 1948. Subject: Inspection Tour, The Fisher Governor Co. Plant, Marshalltown, Ia. Speaker: A. E. Ehrke, assistant to vice president and general sales manager, Fisher Governor Co. Mr. Ehrke gave a very fine discussion on the history of design and development of pressure regulators in an effort to maintain a constant downstream flow, under varying flow, and demand and fluid conditions. This talk was accompanied by slides and blackboard illustrations and was of great interest to those in attendance. Other Features: The nominating committee reported on a slate of officers for the coming year. A report on the heating and air conditioning conference held at Iowa State College on March 22 and 23 was given by Marvin Gould. Attendance 35.

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KANSAS CITY—March 1, 1948. Subject: Experiences in the Air Conditioning Industry. Speaker L. T. Avery, second vice president of ASHVE and president, Avery Engineering Co., Cleveland, Ohio. Problems encountered in connection with penicillin production and radar crystal manufacturing were cited by Mr. Avery. For many of these industries, especially during the war, new methods of air distribution, sterilization and dehumidification were developed. Other Features: Comments on the functioning of the Society and a brief account of the 54th Annual Meeting were given by Mr. Avery. The chapter's delegate to the Annual Meeting, Henry Gould, presented his report on the Meeting. Members of the nominating committee, who will select nominees for 1948 officers, were appointed. Attendance 72.

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MANITOBA—March 18, 1948. Subject: Business Meeting. A letter was read from the Ontario Chapter advising that a substantial surplus from the Canadian Room fund for the Annual Meeting would be carried over and applied to next year's Canadian Room. The secretary was instructed to write a letter expressing the chapter's appreciation to N. W. Kingsland of *Sanitary Age* in appreciation of his efforts to make the Canadian Room a great success. Free use of a film entitled *Water, Water Everywhere*, was offered by the American Cyanamid Co. Following a discussion, the secretary was asked to secure this film for showing at either the September or October meeting. A lengthy discussion was then held regarding a special meeting to be held in April when G. L. Tuve, president of the Society, will be the honored speaker. The most favorable night for the regular monthly meeting was also discussed and a motion was carried that a questionnaire issued to the membership should decide the issue.

A nominating committee was appointed and instructed to present a slate of officers at the following meeting. Attendance 21.

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MEMPHIS—March 15, 1948. Subject: Panel Heating and Its Control. Speaker: Ferdinand Jehle, chief engineer, Hoffman Specialty Co., Indianapolis, Ind. Mr. Jehle presented a fine talk which he illustrated with slides. Other Features: A report of the nominating committee for the election of officers to be held at the April 12th meeting was read. Attendance 41. Attendance ratio 0.91.

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MICHIGAN—April 12, 1948. Subject: A General Discussion on an Interesting Phase of Heating with Forced Hot Water. Speaker: R. E. Moore, vice president, Bell & Gossett Co., Morton Grove, Ill. Mr. Moore's talk, accompanied by slides, covered trends and developments in forced hot water heating. Various types of forced hot water systems which have been applied to different types of building and housing projects during the last several years, were illustrated. E. F. Hyde reported on the arrangements made for the chapter's annual meeting at Meadowbrook Country Club in May. A report was given by E. E. Dubry, the chapter's representative on a committee which has been drawing up a proposed new city ordinance, for the regulation of the operation and inspection of boilers and prime movers and for the licensing of engineers and operators, involved in this work. Attendance 122.

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MINNESOTA—April 5, 1948. Subject: Professional Air Control in 1960. Speaker: G. L. Tuve, president of the ASHVE and professor of mechanical engineering, Case Institute of Technology, Cleveland, Ohio. Professor Tuve outlined the advances made in all phases of engineering up to the present time. The Minnesota Chapter was challenged by Professor Tuve to collaborate with the University of Minnesota and with Rochester doctors to further investigate the relations between comfort and health. ASHVE research fundamentals were outlined as well as plans for future research projects at the Laboratory. Professor Tuve emphasized that the Laboratory is always open to suggestions from members. Other Features: President Tuve presented the Life Membership certificate to H. E. Gerrish, who was one of the founders and organizers of the Minnesota Chapter. C. E. Wiser announced that a National Warm Air Association School will offer the subject of *Theory and Practices of Warm Air Heating*, in Minneapolis April 21 through 23 and will be conducted by G. A. Voorhees, head of the education department of the *National Warm Air Heating and Air Conditioning Association*. Details of the chapter's annual party, to be held on May 12, were outlined by B. F. McLouth. Pres. L. C. Gross discussed the possibility of the Society's holding the 1949 Semi-Annual Meeting in Minneapolis, after a unanimous vote indicated that the chapter was in favor of this proposal. Ralph True of Ducks Unlimited presented the entertainment feature of the evening which consisted of a demonstration of the many various types of duck calls, followed by a colored movie entitled *Bred on the Waters*. Not only the duck hunters in the audience, but also the remainder of those present enjoyed this novel program. Attendance 140.

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MONTREAL—April 12, 1948. Subject: Curing Fire Hazards in Heating, Ventilating and Air Conditioning Systems. Speaker: Mr. McIsaacs,

President G. L. Tuve presents Life Membership certificate to H. E. Gerrish at April meeting of the Minnesota Chapter

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Fire Prevention Department of the City of Montreal. A vote of thanks was given to Mr. McIsaacs for a very interesting talk. Other Features: The meeting was turned over to Leo Garneau, incoming president, by T. H. Worthington, retiring president, who was accorded the appreciation of the chapter for his fine service as president. F. A. Hamlet, discussed the past presidents' pins and requested comments on items which the chapter would like discussed at the Council Meeting, to be held in Cleveland on April 19. Types of meeting papers which the members would like presented at meetings of the Society were discussed. Pres. Leo Garneau announced that the Semi-Annual Meeting would be held on June 22 and 23 at Bretton Woods and 17 members of the chapter indicated that they planned to attend. F. G. Phipps, chairman of the finance committee, reported on the favorable financial standing of the Chapter. Attendance 40.

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NEBRASKA—April 13, 1948. Subject: Air Conditioning and Cooling with Lithium Chloride. Speaker: F. M. Johnson, sales manager, Surface Combustion Corp., Toledo, Ohio. Mr. Johnson's talk was well illustrated and a lively discussion period followed. Other Features: An abstract of the Council Minutes of the meetings held during the Annual Meeting, by A. V. Hutchinson, secretary of the Society, was read by G. W. Colburn. A discussion was held on plans for future meetings and it was announced that emblems will be presented to all past presidents at the June meeting. Attendance 38.

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NEBRASKA—March 2, 1948. Subject: Process Air Conditioning. Speaker: L. T. Avery, second vice president of the ASHVE and president, Avery Engineering Co., Cleveland, Ohio. Mr. Avery prefaced his instructive and interesting talk on process air conditioning with a brief talk on the Society in general and on the HEATING, VENTILATING, AIR CONDITIONING GUIDE. Other Features: Results of a questionnaire on meeting date and types of subjects preferred for meetings were reported by K. R. Magarrell, vice president of the chapter. Attendance 31.

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NEW YORK—March 15, 1948. Subjects: Beauty in Bridges, and The Engineer and His Profession. Speaker: Dr. David B. Steinman, noted bridge builder, author, lecturer and engineer. For his first subject, Beauty in Bridges, colored slides of some of the bridges designed and built by Dr. Steinman were shown. The theme of this talk was that bridges need not be ugly structures but actually can be made beautiful. The Mt. Hope bridge in Rhode Island, which was designed by Dr. Steinman, won an award as the most beautiful bridge in a national contest. After a brief intermission Dr. Steinman spoke on his second subject, which was an appeal for the uniting of the engineering profession as a whole into one *National Society of Professional Engineers* composed of the 90 separate organizations which exist at the present time. The movement toward a national Society started in 1934 and has grown, largely through the efforts of Dr. Steinman, to a group which now totals 17,000 in membership. Dr. Steinman pointed out that while people, firms and groups lay claim to the title of engineer with no validity for such a claim, such a situation does not exist in the medical and legal professions, which are organized into a national body devoted to protecting the welfare of the many groups of which they are composed. Only registered professional engineers in any of the 48 states can belong to the *National Society of Professional Engineers*. Dr. Steinman closed his address with a reiteration of his plea for all engineers to belong not only to their particular technical societies but also to the *National Society of Professional Engineers*. Other Features: Nominations for officers for the coming year were announced by Pres. M. C. Giannini. An auditing committee and a board of tellers were appointed by President Giannini to report at the next meeting. W. H. Driscoll, past president of the Society and of the Chapter, presented a Certificate of Life Membership to Arthur Ritter, a charter member of the New York Chapter. President Giannini announced several meetings under the auspices of the Technical Societies Council of New York. Attendance 60. Attendance ratio 0.14.

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NORTHERN OHIO—March 8, 1948. Subject: Open Forum on Fan Applications. Moderator: Clark Humphreys, senior engineer, ASHVE Research Laboratory. E. F. Morse discussed the standardization of fans and the use of *National Association of Fan Manufacturers'* publications. L. W. Dunbar outlined the history of the development of the various types of fans with particular reference to the radial blade, forward curve and backward curve types. R. E. Sherman discussed propeller fans, and tube and vane axial fans. W. M. Rowe discussed fan systems. Philip Cohen completed the formal part of the program with a brief review of all subjects, with comments. Other Features: L. T. Avery, second vice president of the ASHVE, gave a brief report of his visit to the Kansas City, Nebraska, and Iowa Chapters. Cyril Tasker, director of research at ASHVE Research Laboratory, gave a short talk on his recent trip to Washington, D.C., and Atlanta, Ga. W. R. Moore, chapter delegate to the 54th Annual Meeting presented his official report of the meeting. Attendance 55.

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NORTH TEXAS—April 12, 1948. Subject: Axial Flow Fans. Speaker: W. R. Heath, chief engineer, Buffalo Forge Co., Buffalo, N. Y. Mr. Heath's informative talk covered the application and principle of design of axial flow fans. Other Features: A report was given by John Ray, for the program committee and A. B. Ullrich, chapter treasurer, reported on chapter finances. C. Rollins Gardner gave a brief report on the chapter delegates meeting held during the Annual Meeting in New York. Attendance 56.

OKLAHOMA—April 12, 1948. Subject: Water Cooling Tower Selections for Refrigeration and Air Conditioning. Speaker: Howard E. Degler, technical director, Marley Co., Inc., Kansas City, Kans. Mr. Degler covered his subject thoroughly, particularly with reference to local conditions and problems. Other Features: A letter from W. M. Wallace, II, Durham, N. C., was read, suggesting names of individuals who might be considered by the National Nominating Committee as nominees for officers in the Society. Guests from the University of Oklahoma and the Oklahoma A. & M. College were introduced. Attendance 48. Attendance ratio 0.39.

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OKLAHOMA—March 8, 1948. Subject: Baseboard Heating. Speaker: D. Bennett, manager, unit heater and radiation department, Warren Webster & Co. Mr. Bennett's presentation was well given and extremely well accepted. An open discussion of the merits of baseboard heating was then held. Other Features: J. R. Patten reported for the membership committee and Howard Shafer as chairman of the program committee announced the speaker for the next meeting. Attendance 35. Attendance ratio 0.33.

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OREGON—April 2, 1948. Subject: Electronics in Peace. Speaker: Gordon Volkenant, research engineer, Minneapolis-Honeywell Regulator Co. A highly informative and interesting illustrated lecture on the war-time development and uses of electronic devices and what their development means to present-day living. Other Features: This was an informal annual meeting and banquet session and a resume of the past year was given by C. W. Brissenden, president. A presentation was made of statuary bronze book-ends to retiring president Brissenden. Attendance 260.

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PHILADELPHIA—April 8, 1948. Subject: Centrifugal Pumps as Applied to Heating and Air Conditioning. Colored motion pictures were shown describing modern methods used in mining lead ore. Both the talk and pictures were enjoyed by the audience. Other Features: Edwin Elliot, chairman of the auditing committee reported that the treasurer's books were audited and found to be in proper order. The following were unanimously elected officers for the coming year: J. O. Kirkbride, *president;* F. H. Buzzard, *first vice president;* J. W. McElgin, *second vice president;* E. K. Wagner, *treasurer;* M. E. Barnard, *secretary;* and C. F. Dietz, member of the *board of governors.* Attendance 80.

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PHILADELPHIA—March 11, 1948. Subject: Internal Chemical Conditioning for Recirculated Water. Speaker: C. M. Sterne, chief engineer, Metropolitan Refining Co. The talk was well received by the membership with a lively discussion period following. Other Features: The minutes of the February 12th meeting and the treasurer's report were read and approved. Attendance 71.

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PITTSBURGH—April 12, 1948. Subject: Condensation in Building Construction. Speaker: H. E. Robinson, National Bureau of Standards, Washington, D. C. Mr. Robinson explained that due to building conditions in general and tighter construction with more efficient insulation, condensation can happen and will happen in some cases. Slides of construction and heat transfer were used to indicate where the insulation in walls should be. Mr. Robinson concluded his presentation with a description of a housing project in Washington, D. C., and how he solved the condensation problem in this case. Other Features: The nominating committee report on officers for the coming year was given by F. C. McIntosh. Pres. D. W. Loucks reported that the new chapter rosters were ready for mailing and E. C. Hach reported for the membership committee. J. L. McCann spoke on a golf meet scheduled for May 25 for which L. S. Walker will donate the winning cup. Attendance 70.

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ROCKY MOUNTAIN—March 3, 1948. Subject: Analysis of Fluid Flow. Speaker: Walter Leaf, research engineer, D&RGW Railroad, Denver, Colo. Mr. Leaf discussed primarily the flow of products of combustion in locomotives, and how studies have been made on this subject. Many of his company's operating problems including cinder cutting, blower performance and tunnel ventilation have been overcome by such studies. Other Features: H. H. Herman, delegate to the 54th Annual Meeting of the Society in New York, gave his report on the Meeting. The nominating committee presented a slate of officers for the coming year and it was announced that election of officers will be held on April 7. An announcement was made that an Indoor Comfort Conference conducted by G. A. Voorhees and sponsored by the *National Warm Air Heating & Air Conditioning Association* will be held in Denver on May 5, 6 and 7. Attendance 21.

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ST. LOUIS—April 6, 1948. Subject: The District Heating Industry. Speaker: R. M. McQuitty, steam utilization engineer, Union Electric Co. of Missouri. Mr. McQuitty began his talk with the origin of the district system in the East and outlined the progress up to the present time of the various district systems. His unusual, interesting discussion was illustrated with slides. Other Features: Recommendations for officers for the coming year were given by the nominating committee. After the chapter members indicated their approval, a committee was appointed to make plans for a spring dinner dance. Among the new members introduced was J. H. Van Alsburg, formerly of the Michigan Chapter, who stated that, since St. Louis was to be his permanent address, he intended to join the chapter. Attendance 53.

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ST. LOUIS—February 2, 1948. Subject: Thermostatically Controlled Warm Air Registers in Forced Air Heating. Speaker: Bruce McLouth, consulting engineer, Dole Valve Co. Mr. McLouth, in his interesting review, gave a history of warm air heating from the early days of

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mankind when campfires were used to the present automatic heating plants. Other Features: It was unanimously passed by the chapter that two new committees be appointed, an Advancement and Admission Committee, and a Guide and Research Committee. Pres. J. H. Carter, after discussing the cooperation of the ASHVE with the Engineers Club relative to their new building, announced that W. A. Russell had been appointed delegate to meet with the various engineering societies and J. S. Rosebrough, alternate. A motion was approved that past president emblems be furnished to all local past presidents. A nominating committee to select nominees for 1948 officers was appointed. The secretary, J. S. Rosebrough, read an invitation from the *National Society of Corrosion Engineers* to attend their Third Annual Convention to be held April 4-8, 1948, at the Jefferson Hotel, St. Louis. D. M. Allen, a member of the Kansas City chapter, and of the ASHVE Council, spoke briefly on a Council meeting held in New York. Attendance 58.

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SOUTH TEXAS—March 19, 1948. Subject: Benefits of Vibration Control as Applied to Heating, Ventilating and Air Conditioning Equipment. Speaker: S. Rosenzweig, president, Korfund Co., Inc., New York, N. Y. A question and answer period followed Mr. Rosenzweig's presentation. Other Features: A. F. Barnes, delegate to the Engineers' Council, reported on progress being made on a city code. The 4th Annual Air Conditioning Conference to be held at the Texas A. & M. College on April 26, 27 and 28 was announced by B. P. Fisher. A lengthy report on the chapter delegates committee meetings held during the 54th Annual Meeting was given by L. L. Ladewig, delegate of the South Texas Chapter. T. V. Burns of the Texas A. & M. Student Branch also reported on the Annual Meeting and S. E. Ammons, president of the Student Branch, thanked members for their contributions which made it possible for three student members to attend the meeting. Pres. D. M. Mills announced that past presidents' pins had been ordered for all past presidents. Attendance 92.

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SOUTHERN CALIFORNIA—March 17, 1948. Subject: Reaction of Plant Growth in Air Conditioned Green Houses. Speaker: Dr. I. W. Went, plant physiologist and biologist, California Institute of Technology. Dr. Went's research and experiments over the past several years have proved that controlled climatic conditions, light, and carbon dioxide are very important in the development of plant life. For instance, he has found that day temperatures have very little effect on the growth of tomatoes but that a controlled night temperature of 65 deg resulted in rapid growth and, the longer the dark period, the sooner the plant would fruit. Dr. Went pointed out that much more research must be done to find the proper conditions for various plants and that air conditioning is an all important factor in this field. Attendance 63.

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WASHINGTON—March 19, 1948. *Ladies Night*. Subject: Hayseed Electronics. Speaker: Gordon Volkenant, Minneapolis-Honeywell Regulator Co. Mr. Volkenant told of the part played by electronics during the recent war and gave many interesting facts concerning America's recently developed fighting equipment. Of particular interest to the ladies was Mr. Volkenant's reference to electronic ranges which will sizzle a steak in 10 seconds. Other Features: Nominations for 1948 chapter officers were announced by L. Bert Nye, chairman of the nominating committee. All other chapter business was dispensed with, out of deference to the ladies present. Attendance 80.

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WESTERN MICHIGAN—March 24, 1948. Subject: Electronics. Speaker: Gordon Volkenant, physicist, Minneapolis-Honeywell Regulator Co. Mr. Volkenant gave a most interesting demonstration and lecture on electronics pertaining to all phases of application. Other Features: A resolution was read by F. R. Bishop regarding the death of E. C. Henry and a copy was sent to Mrs. Henry. This was a joint banquet meeting of the Western Michigan Chapter of the ASHVE and the student body of the 17th Annual Forced Warm Air Conference, Michigan State College, East Lansing, Mich. Dean Dark, of Michigan State gave a few words of welcome and also outlined briefly some of the outstanding features of the college. A review of the history of the Warm Air Conferences from their inception in 1930 was given by B. F. McLouth. Attendance 335. Attendance ratio 0.94.

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WISCONSIN—March 17, 1948. Joint Meeting of *ESM*, ASHVE and *ASRE*. Subject: Heating by Refrigeration—The Heat Pump. Speaker: W. Parkerson, rate engineer, Middle West Service Co., Chicago, Ill. Mr. Parkerson traced the history of the heat pump and described its basic principles and limitations. He also commented on present installations. Other Features: Since this was a joint meeting all regular business was dispensed with. Attendance 62.

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Student Branch

TEXAS A. & M.—March 18, 1948. Subject: Sales Engineering. Speaker: Harold Broadwell, application engineer, York Corp., Houston, Tex. Mr. Broadwell discussed the field of sales engineering and mentioned various prerequisites for success in this field. Other Features: B. P. Fisher of the South Texas Chapter extended an invitation to the students to attend a South Texas Chapter meeting to be held on March 19 and also to participate in an air conditioning short course to be held during the latter part of April. A report of the activities of the Student Branch was requested by Mr. Fisher for publication in the South Texas Chapter bulletin, the *Hot Air Recorder*. Attendance 50.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When the replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month, 152 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 25, 1948, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

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RAUCH, M. A., Mech. Engr., Office of Technical Service, U. S. Public Health. PROPOSERS: H. H. Hill, H. W. Redmile. SECONDERS: F. M. Thuney, L. F. Flagg.

REED, J. H., Gen. Sales Mgr., The Dykes Co. Inc. PROPOSERS: O. J. Dykes, Jr., W. E. FitzGerald. SECONDERS: B. Segall, Jr., L. D. Parsons, Jr.

RIGBY, C. J., Sales Engr., Air Cond. Dept., General Electric Co., PROPOSERS: S. A. Durbin, J. P. Turner, Jr. SECONDERS: F. H. Faust, R. U. Berry.

ROBINSON, W. M., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

ROGERS, CHARLES, Prop., Chas. Rogers. PROPOSERS: F. R. Johnson, B. W. Farnes. SECONDERS: R. W. Martig, W. R. Norte.

ROUSSEAU, J. A., Owner, Coast Heating & Sheet Metal. PROPOSERS: T. H. McClung, A. C. Clark. SECONDERS: A. K. Johnston, E. R. Lokey.

RUF, C. E., Co-op Student, Frigidaire Division of General Motors Corp. PROPOSERS: C. F. Wood, Herman Worsham. SECONDERS: L. B. Wocholski*, R. H. Stanley*.

RUSH, C. B., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

ST. GERMAIN, W .E., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

SASSER, R. E., Mgr. Industrial Insul. Div., Alexander-Stafford Corp. PROPOSERS: T. D. Stafford, S. H. Downs. SECONDERS: O. D. Marshall, V. C. Dean.

SAWYER, H. W., Sales Engr., Minneapolis-Honeywell Regulator Co. PROPOSERS: E. J. Anderson, E. F. Hyde. SECONDERS: W. P. Ziemba, A. E. Knibb.

SCHAEFER, A. J., Air Cond. Engr., Firestone Tire & Rubber Co. PROPOSERS: L. E. Slawson, J. E. Wilhelm. SECONDERS: L. S. Oswald, L. C. Burkes.

SCHNEIDER, C. R., Sales Engr., Dealer's Supply Co. PROPOSERS: W. W. Wise, C. B. Cole. SECONDERS: G. G. Bailey, L. J. O'Callaghan.

SCHUBERT, J. D., Application Engr., Minneapolis-Honeywell Regulator Co. PROPOSERS: J. H. Spaan, W. B. Ash. SECONDERS: F. X. Loeffler, Jr., C. Y. Trowbridge.

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SHAFFNER, C. R., JR., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

SMITH, C. M., JR., Sales Engr., Martin-Johnson Engrg. Co. PROPOSERS: G. H. Meffert, E. T. Gessell. SECONDERS: R. G. Luford, J. C. Chase.

SMITH, J. B., Sales Engr., Johnson Service Co. PROPOSERS: P. R. Babcock, F. W. Hook. SECONDERS: Karl Baldwin, Jr., F. J. White.

SMITH, M. A., Owner, Marvin A. Smith Co. PROPOSERS: A. F. Avera, C. C. Young. SECONDERS: M. L. Brown, B. G. Mercer.

SOLON, J. A., Controller-Asst. Treas., McDonnell & Miller Inc. PROPOSERS: E. N. McDonnell, J. W. James. SECONDERS: N. W. Swanson, George La Roi.

STATEN, F. A., Secy.-Treas., Acme Metal Works, Inc. PROPOSERS: T. H. McClung, V. A. Pfeifer. SECONDERS: E. R. Lokey, R. P. Lankow.

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STRONG, K. M., Student, Michigan College of Mining & Technology. PROPOSERS: F. E. Wittig, R. A. Fricke. SECONDERS: E. E. Hophemeyer, Jr.*, A. P. Young*.

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SYKES, W. A . JR., Student, North Carolina State College of Agriculture & Engineering. PROPOSERS: L. L. Vaughan, R. B. Rice. SECONDERS: K. P. Hanson*, N. W. Conner*.

TARNOFF, E. M., Prod. Dept., Prat-Daniel Corp. PROPOSERS: G. F. Begoon, E. H. R. Pegg. SECONDERS: C. H. Pinches*, A. A. Petersen*.

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THOMSON, R. C., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

THORNBERRY H. R., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

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TRUAX, J. W., Design Engr., Columbia Specialty Co., Inc. PROPOSERS: Henry Latterman, Jr., B. P. Robeson. SECONDERS: P. H. McKim, J. G. Werner.

TWIETMEYER, H. E., Res. Engr., Res. Labs. Div. General Motors Corp. PROPOSERS: W. N. Witheridge, L. A. Burch. SECONDERS: N. B. Elliott, C. F. Huddle*.

URSINI, J. V., Owner, James V. Ursini Co. PROPOSERS: C. J. Wahnquist, Jr., Winfield Roeder. SECONDERS: J. J. Moore, D. M. Hummel.

VINSONHALER, C. I., Sales Engr., Geo. E. Mead Co. PROPOSERS: D. O. Mead, R. M. Towne. SECONDERS: R. E. Le Riche, J. D. Sparks.

WALLACE, J. W., JR., Sales, American Radiator & Standard Sanitary Corp. PROPOSERS: W. R. Norte, R. H. Liniger. SECONDERS: J. A. Beers, R. C. Chewning.

WALLBILLICH, J. J., JR., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

WEIRICH, H. P., Sales & Design, Minneapolis-Honeywell Regulator Co. PROPOSERS: J. E. Marshall, C. E. Bentley. SECONDERS: J. E. Murray, S. W. Terry.

WILSON, R. W., Student, Purdue University. PROPOSERS: F. B. Morse, Y. S. Touloukian. SECONDERS: W. E. Fontaine*, W. T. Miller.

WINTER, M. D., Sales Engr., M. O. Cox Sheet Metal Co. PROPOSERS: T. H. McClung, E. R. Lokey. SECONDERS: C. W. Brissenden, A. C. Clark.

WISEHART, M. J., Pres., Wisehart Heating & Sheet Metal, Inc. PROPOSERS: C. F. A. Locke, J. W. Ely. SECONDERS: R. A. Donie, J. F. Payton.

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by **Art. B-III**, *Section* 8, of the By-Laws, the following list of candidates elected:

Members

EIGENMANN, ALBERT, Cons. Engr., Davos-Dorf, Switzerland.

ELLEMBERGER, F. R., Section Head, Bloomfield, N. J.

ERICKSON, E. A., Res. Engr., Columbus, Ohio.

FITZGERALD, J. J., Engr., Waterbury, Conn.

FITZGERALD, W. E., Pres. & Gen. Mgr., Shreveport, La. (*Advancement*)

GREEN, P. A., Engr., Philadelphia, Pa.

HELLER, M. MENDELL, Engr., Shreveport, La.

HENDERSON, F. P., Dist. Engr., Oakland, Calif.

KOETZ, LESTER, Partner, Zion, Ill. (*Reinstatement*)

LIEBERG, O. S., Partner, Pasadena, Calif.

LUDWIG, N. C., Res. Engr., Buffington, Indiana.

MEADE, A. J., Prod. Mgr., Pittsburgh, Pa.

MEININGER, C. W., Development Engr., New Kensington, Pa.

MILES, H. W., Des. Engr., Toronto, Ont., Canada.

MURRAY, G. F. M., Gen. Mgr., London, England.

MUSGRAVE, B. LE B., Vice Pres. & Dir., New York, N. Y.

NOSS, O. F., Asst. Sales Mgr., New York, N. Y.

PAULSON, W. D. JR., Disr. Repr., Oakland, Calif.

PHELPS, K. G., Mech. Engr., So. Lyon, Mich.

SASSER, J. F., Chief Engr., Pittsburgh, Pa.

SCOTT, A. F., Dallas, Tex. (*Reinstatement*)

Associate Members

AUSTIN, H. H., Supt. Installation & Serv. Dept., Shreveport, La.

BIRCH, F. W., Mgr., Toronto, Ont., Canada.

BOEHLER, L. E., Engr., Milwaukee, Wis.

BOGGS, DENNIS M., Service & Installation Engr., Shreveport, La.

DAVIS, F. H., Owner, Buffalo, N. Y.

FREE, WALTER G., Pres., Shreveport, La.

HAMMOND, W. A., Pres., Buffalo, N. Y.

HARWOOD, VAN NESS, Sales, Buffalo, N. Y.

HOPPER, R. W., Installation & Serv. Mgr., Shreveport, La.

JOHNSON, R. M., Merchandising Supervisor, Beverly Hills, Calif.

JOHNSON, T. V., Assoc. Ed., Chicago, Ill.

KEY, LACY, JR., Jr. Engr., Houston, Tex.

L'ESPERANCE, J. R. L., Mgr., Owner, Montreal, P. Q., Canada.

LOKKEN, C. A., Engr., La Crosse, Wis.

MAY, A. G., Estimator, Knoxville, Tenn.

MILLER, J. W., Sales, Dallas, Tex.

MITCHELL, J. D., Sales Engr., Oakland, Calif.

OSTIGUY, G. L., Montreal, P. Q., Canada.

PETTI, O. A., Engr., Detroit, Mich.

PRATT, R. E., Instructor, Chicago, Ill.

RILEY, W. E., SR., Sales Engr., Washington, D. C.

ROBINSON, W. J., Sales, Montreal, P. Q., Canada.

ROONEY, L. H., Sales, New York, N. Y.

SCOGGINS, R. M., Mgr., Shreveport, La.

SHAPIRO, PAUL, Kansas City, Mo.

SHEILS, J. S., Htg. & Vent. Estimator, New York, N. Y.

SHUMAN, DON W., Mgr. Refrig. & Air Cond. Dept., Charlotte, N. C.

SILVA, J. V. B., Mgr. Air Cond. Dept., Lisboa, Portugal.

SNELLER, A. J., Sales Engr., Kansas City, Mo.

SODERBERG, C. H., Dist. Mgr., Philadelphia, Pa.

TACK, P. B., Sales Engr., Seattle, Wash.

TEMPEL, W. W., Mgr., Tel-Aviv, Palestine.

WEINERT, F. C., Sec'y, Detroit, Mich. (*Reinstatement*)

WHEELER, W. R., Mech. Engr., Baltimore, Md.

WOOLLEY, N. L., Chief Engr., Dayton, Ohio.

Junior Members

BALLARD, I. MC G., Air. Cond. Engr., Shreveport, La.

BENNETT, L. M., Mech. Engr., Toronto, Ont., Canada.

BUCHANAN, WILLIAM M., 3rd Engr., Shreveport, La.

DOBRIN, R. L., Mech. Engr., Cleveland, Ohio.

FIDELI, F. A., Sales Engr., New York, N. Y.

GAWMAN, D. S., Chief Tech. Advisor, New York, N. Y.

GROSS, H. E., Partner, Philadelphia, Pa.

HULTZ, W. S., Htg. Engr. & Partner, Elyria, Ohio.

KNIGHT, J. L., Toronto, Ont., Canada.

LINDVALL, D. E., Dir. House Heat, Rockford, Ill.

LOVE, N. M., Mech. Draftsman, Atlanta, Ga.

MARTELLI, R. V. B., Jr., Engr., Detroit, Mich.

POLIVKA, N. L., Mech. Engr., La Grange, Ill.

SMITH, IRWIN, Draftsman, New York, N. Y.

WHELLER, H. S., JR., Prod. & Cost Control Engr., New York, N. Y.

Student Members

BUENZ, N. F., Student, A. & M. College of Texas, College Station, Tex.

CHAPMAN, H. G., JR., Student, David Ranken Jr. School of Mechanical Trades, St. Louis, Mo.

DICKSON, J. A., JR., Student, A. & M. College of Texas, College Station, Tex.

FREEMAN, G. D., Student, Michigan College of Mining & Technology, Houghton, Mich.

FRITCHMAN, J. F., Student, Purdue University, W. Lafayette, Ind.

HENRIKSEN, C. K., Student, Purdue University, W. Lafayette, Ind.

HOWARD, C. P., Student, A. & M. College of Texas, College Station, Tex.
KARIBO, R. J., Student, David Ranken Jr. School of Mechanical Trades, St. Louis, Mo.
KENNEDY, B. M., Student, A. & M. College of Texas, College Station, Tex.
KILPATRICK, R. S., Student, Michigan College of Mining & Technology, Houghton, Mich.
KLION, SANFORD, Student, Massachusetts Institute of Technology, Cambridge, Mass.
KNEPPER, R. F., JR., Student, A. & M. College of Texas, College Station, Tex.
LYNCH, D. T., Student, Michigan College of Mining & Technology, Houghton, Mich.
MARTIN, M. B., JR., Student, A. & M. College of Texas, College Station, Tex.
MAYO, J. T., Student, A. & M. College of Texas, College Station, Tex.
NANCE, J. W., Student, A. & M. College of Texas, College Station, Tex.
PAPACH, G. M., Student, Purdue University, W. Lafayette, Ind.
SHIH, CHENG, Student, University of Illinois, Urbana, Ill.
SIENER, G. J., Student, Purdue University, W. Lafayette, Ind.
SMITH, T. L., JR., Student, Washington University, St. Louis, Mo.
STREETMAN, N. V., JR., Student, A. & M. College of Texas, College Station, Tex.
STURGELL, J. W., JR., Student, David Ranken Jr. School of Mechanical Trades, St. Louis, Mo.
TAYLOR, ROBERT W., Student, W. Lafayette, Ind.
THOMPSON, J. T., Student, Michigan College of Mining & Technology, Houghton, Mich.
TRENARY, BRYANT, Student, Purdue University, W. Lafayette, Ind.
VAN RAVENSWAAY, R. C., Student, Massachusetts Institute of Technology, Cambridge, Mass.
WALKER, R. A., Student, Southern Methodist University, Dallas, Tex.
WHITACRE, J. A., JR., Student, A. & M. College of Texas, College Station, Tex.
WHITE, J. B. Student, Purdue Univerty, W. Lafayette, Ind.
WISTI, W. M., Student, Michigan College of Mining & Technology, Houghton, Mich.

P. S. CHASE DIES SUDDENLY

The sudden death of Peter S. Chase, owner, Chase Co., Eugene, Ore., has been announced. Mr. Chase had been owner of his own plumbing, heating and air conditioning firm since 1921.

A native of Dufur, Ore., Mr. Chase was born on March 25, 1890. He served in the U. S. Marine Corps prior to attending the University of Pennsylvania, Philadelphia, Pa. Among the buildings for which he designed and engineered the heating, plumbing and ventilating installations are the Oregon hotel, Toledo hospital, Douglas County Court House and the Drain Schools.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS join his associates in the Oregon chapter in extending their deepest sympathy to Mrs. Chase and his family.

JOHN H. FAULKNER DIES

The death of John H. Faulkner, owner and manager, John H. Faulkner Co., Seattle, Wash., on March 1, 1948, has been announced. He was elected a member of the ASHVE in 1941.

Mr. Faulkner was born 51 years ago at Seattle, Wash. He received his early engineering training at the University of Washington. One of his first positions was office manager of McMahon Plumbing & Heating Co. In 1924 he became an assistant steam boiler inspector for the city of Seattle. He joined the staff of the Birchfield Boiler Corp., Tacoma, Wash., in 1926, leaving two years later to act as a sales engineer for The Trane Co. His next position was as secretary-treasurer of the Langdon-Faulkner Co., Inc., sales engineers of heating, ventilating and air conditioning equipment, a position which he held until 1942 when he became a partner of this firm. He organized his own company under the name of the John H. Faulkner Co., in 1945.

To his survivors, the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sympathy for their untimely loss.

F. W. HANBURGER DIES AT 74

Fred W. Hanburger, seventy four, an instructor of civil engineering at the College of the City of New York, died at his home on March 17, 1948. Mr. Hanburger had been a member of the Society for 14 years.

Born at Ashland, Pa., on January 1, 1874, Mr. Hanburger received his education at Ashland public schools. In 1892 he joined the engineering department of the Philadelphia & Reading Coal & Iron Co., where he was engaged in surveying, mapping and general design work. Later, he made mine surveys involving mine ventilating and water supply for the same company. His next position was with the Rosano Mining Co., Honduras, Central America, as an engineer of mines. He returned to the United States during 1900 and accepted a position as assistant general superintendent for the Westmoreland Coal Co., Irwin, Pa. After two years with this company he left in order to become a consulting engineer. He first joined the faculty of City College as an instructor in 1916, teaching military mapping and surveying for the United States government. Following the first world war, he remained in government service until 1920 when he returned to the faculty of City College as an instructor of heating and ventilating. During his many years with the City College he had also been actively engaged in consulting work.

Surviving Mr. Hanburger are his widow, Mrs. Lulu Howe Hanburger, and his son, Lieutenant Colonel Christian Hanburger, of the U. S. Army Engineers, to whom the Officers and Council of the ASHVE offer their sincere condolences.

DEATH OF T. J. HESTER

On March 4, 1948, Thomas J. Hester, president of Hester-Bradley Co., St. Louis, succumbed to pneumonia after a month's illness. Mr. Hester had been a member of the ASHVE since 1919.

A native of St. Louis, Mr. Hester was born on March 1, 1878, and received his education in St. Louis public schools preparatory to attending Christian Brothers College. His early experience was obtained at Hester-Bradley Co. and his shop experience with Oakill Collins and Co. In 1897 he joined the staff of the L. M. Rumsey Mfg. Co., leaving 10 years later for a position with J. B. Clow & Sons, Chicago, Ill., where he remained three years. In 1910 he became one of the organizers and an officer of the Iroquois Engineering Co., where he

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T. J. Hester
St. Louis, Mo.

was in charge of the St. Louis office. Hester-Bradley Co., which was formed by the partnership of E. P. Bradley and Mr. Hester in 1914, is the St. Louis representative for Warren Webster & Co., Nash Engineering Co., and Kewanee Boiler Corp.

Mr. Hester was a member of the Knights of Columbus, a sponsor of the St. Louis Municipal Opera and a member of the Missouri Athletic Club. He served the Athletic Club as secretary for a year and was a member of the board of governors.

Surviving Mr. Hester are his widow, two daughters, Mrs. Ruth Harvey and Mrs. Elizabeth Hibbs, and two sisters, Mrs. Nellie Mullen and Miss Tessie C. Hester, to whom the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincere sympathy.

DEATH OF E. W. KLEIN, LIFE MEMBER

The death of Edward W. Klein, recently elected Life Member of the ASHVE, has been announced. Death occurred on April 21, 1948, in a private hospital after an extended illness.

Mr. Klein, who was born on August 9, 1882, in Minneapolis, Minn., received his early education at Boys High School, Atlanta, Ga. His B.S. in mechanical engineering was received in 1902 from Georgia School of Technology, where he was a member of Phi Kappa Phi, honorary scholastic fraternity. From 1902 until 1906, Mr. Klein held a position as draftsman with several firms. For the next ten years he was employed by the Engelhart Heating Co., Atlanta, Ga., as a heating engineer. After this position, and for the 30 years prior to his death, Mr. Klein was a steam heating engineer representing various companies in the southeastern states. He had represented Warren Webster & Co. for the past 25 years.

A member of the ASHVE since 1917, Mr. Klein was a charter member of the Atlanta Chapter, and had served as its first president. Mr. Klein, who had also served as president of the local chapter of the *ASME*, was a member of the Pi Tau Sigma engineering fraternity and the *Georgia Engineers Society*, in addition to belonging to several prominent local civic organizations.

His survivors include his widow, Mrs. Berma Daniell Klein; two daughters, Mrs. Allen Webb and Mrs. Lloyd Davis; and two sons, Robert D. Klein and Edward Klein, Jr., to whom the Officers and Council of the Society extend their sincere sympathy.

OTTO WECHSBERG DIES IN NASSAU

Otto Wechsberg, president and general manager, Coppus Engineering Corp., Worcester, Mass., died unexpectedly on March 4, 1948, in Nassau, B.W.I., only four days after his arrival for a month's vacation.

A native of Friedland, Czechoslovakia, Mr. Wechsberg was born on May 25, 1881. He received his mechanical engineering degree at the technical university, Vienna, Austria, in 1906. After arriving in the United States in 1914, Mr. Wechsberg's first position was with Goulds Mfg. Co., Seneca Falls, N. Y., as a designer of centrifugal pumps. In 1915, he joined the staff of the Southwark Foundry, Philadelphia, Pa., where he was also employed as a designer of centrifugal pumps. One year later he joined the Advance Pump & Compressor Co., Battle Creek, Mich., as chief engineer. He became affiliated with the Coppus Engineering and Equipment Co. in 1917. This company and the New England Brass Foundry Co. were consolidated in 1922 to form the Coppus Engineering Co. Mr. Wechsberg became president and general manager in 1926, serving in this capacity until his death.

A member of the ASHVE since 1932, Mr. Wechsberg also held membership in the *ASME*. He is the author of technical papers which he presented at Society meetings.

Those surviving Mr. Wechsberg are his mother, Mrs. Rudolphine Wechsberg of Worcester and a brother Philip, who resides in Czechoslovakia, to whom the Officers and Council of the Society extend their sympathy for their unexpected loss.

LIFE MEMBER DIES IN PHILADELPHIA

The death of James T. J. Mellon, a Life Member of the Society since 1945, and president of the Mellon Co., Philadelphia, Pa., has been announced.

Born on July 3, 1880, in Philadelphia, Pa., Mr. Mellon attended local schools. He received his early drafting experience with the Fawcett Construction Co., Keystone Telephone Co., and S. Faith & Co. He became a member of the Walters, Purks and Mellon firm, contracting engineers, nearly forty years ago. This firm was reorganized as the Mellon Company in 1924.

Mr. Mellon, a member of the ASHVE since 1911, was active in the organization of the Society's Philadelphia Chapter.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their deepest sympathy to his survivors.

DEATH OF ANDREW SHERET

The Society was recently notified of the death, on September 12, 1947, of Andrew Sheret, president, Andrew Sheret, Ltd., Victoria, B.C., and Life Member of the ASHVE.

Mr. Sheret was born 78 years ago in Montrose, Scotland. After his early engineering training he formed his own firm in 1892, and was active in the installation of steam and hot water heating systems. Later, the firm became a representative for various radiator, boiler and plumbing goods firms.

After joining the ASHVE in 1925 as an associate, Mr. Sheret was advanced to full membership in 1929, and was later honored by election to Life Membership.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their deepest sympathy to his survivors.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Headquarters: 51 Madison Ave., New York 10, N. Y. (Tel.: MUrray Hill 3-0291)

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Secretary.....A. V. Hutchinson
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Central New York: Organized 1944. Headquarters, Syracuse, N. Y. President, D. L. Mills, 1407 N. Madison St., Rome. Secretary, J. H. Carpenter, 3515 Midland Ave., Syracuse 7.

Central Ohio: Organized, 1944. Headquarters, Columbus, Ohio. Meets, Third Monday. President, W. M. Myler, Jr., P.O. Box 267, Columbus 16. Secretary, H. G. Hays, 851 W. Third Ave., Columbus 8.

Cincinnati: Organized 1932. Headquarters, Cincinnati, Ohio. Meets First Tuesday. President, H. K. Jennings, 1639 Union Trust Bldg., Cincinnati 2. Secretary, K. B. Little, 832 Temple Bar Bldg., Cincinnati 2.

Connecticut: Organized, 1940. Headquarters, New Haven, Conn. President, E. C. Marsden, 44 Hicks St., Hartford. Secretary, T. L. Arnold, 26 Francis Ave., Hartford.

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Delta: Organized, 1939. Headquarters, New Orleans, La. Meets, Second Tuesday. President, J. S. Burke, 317 Baronne St., New Orleans 9. Secretary, R. B. Guest, 827 Dryades St., New Orleans 13.

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Golden Gate: Organized, 1937. Headquarters, San Francisco, Calif. Meets, First Thursday. President H. V. Hickman, 1129 Folsom St., San Francisco 1. Secretary, R. C. Cushing, 1136 Howard St., San Francisco 3.

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Illinois: Organized, 1906. Headquarters, Chicago, Ill. Meets, Second Monday. President, C. M. Burnam, Jr., 6 N. Michigan Ave., Chicago 2. Secretary, G. W. Bornquist, 629 W. Washington Blvd., Chicago 6.

Indiana: Organized, 1943. Headquarters, Indianapolis, Ind. Meets, Fourth Friday. President, P. R. Jordan, 311 E. South St., Indianapolis 4. Secretary, C. F. A. Locke, 1007 N. Meridian St., Indianapolis 4.

Iowa: Organized, 1940. Headquarters, Des Moines, Ia. Meets, Second Tuesday. President, D. C. Murphy, 840 Fifth Ave., Des Moines 14. Secretary, R. S. Stover, Kresge Bldg., Marshalltown.

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Kansas City: Organized, 1917. Headquarters, Kansas City, Mo. Meets, First Monday. President, P. C. Leffel, 3323 Main St., Kansas City 2. Secretary, Henry Nottberg, Jr., 914 Campbell, Kansas City.

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Manitoba: Organized, 1935. Headquarters, Winnipeg, Man., Canada. Meets, Third Thursday. President, J. R. Stephenson, 278 Main St., Winnipeg. Secretary, D. F. Michie, 278 Main St., Winnipeg.

Massachusetts: Organized, 1912. Headquarters, Boston, Mass. Meets, Third Tuesday. President, R. T. Kern, 51 Claflin St., Leominster. Secretary, D. W. Blair, 59 Frost St., Cambridge.

Memphis: Organized, 1944. Headquarters, Memphis, Tenn. Meets, First Monday. President, E. E. Scott, 918 Dermon Bldg., Memphis 3. Secretary, A. T. Bevil, 1521 Waverly Ave., Memphis 6.

Michigan: Organized, 1916. Headquarters, Detroit, Mich. Meets, First Monday after 10th of month. President, E. F. Hyde, 708 Oakland Ave., Birmingham. Secretary, J. H. Spurgeon, 5050 Joy Rd., Detroit 4.

Minnesota: Organized, 1918. Headquarters, Minneapolis, Minn. Meets, First Monday. President, L. C. Gross, 5324 Oaklawn Ave., Minneapolis 10. Secretary, L. J. Krause, 2753 4th Avenue S., Minneapolis 8.

Montreal: Organized, 1936. Headquarters, Montreal, Que., Canada. Meets, Third Monday. President, Leo Garneau, 25 Salisbury Rd., Pointe Claire, Que. Secretary, R. R. Noyes, 630 Dorchester St. W., Montreal.

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Nebraska: Organized, 1940. Headquarters, Omaha, Neb. Meets, Second Tuesday. President, Verne Simmonds, 2416 Vane St., Omaha. Secretary, G. W. Colburn, 106 N. 15th St., Omaha 2.

New York: Organized, 1911. Headquarters, New York, N. Y. Meets, Third Monday. President, A. A. Bearman, 444 West 56th St., New York 19. Secretary, Carl H. Flink, Room 3000, 51 Madison Ave., New York 10.

North Carolina: Organized, 1939. Headquarters, Durham, N. C. Meets, Quarterly. President, E. S. deWitt, 1211 Commercial Bank Bldg., Charlotte. Secretary, R. M. Warren, Jr., 1001 N. Church St., Charlotte 1.

North Texas: Organized, 1938. Headquarters, Dallas, Tex. Meets, Third Monday. President, B. S. Foss, Jr., 711 Construction Bldg., Dallas 1. Secretary, Herman Blum, Jr., 2812 Fairmount St., Dallas 4.

Northern Ohio: Organized, 1916. Headquarters, Cleveland, Ohio. Meets, Second Monday. President, John James, 3170 W. 106th St., Cleveland 11. Secretary, W. M. Rowe, 1302 Swetland Bldg. Cleveland 15.

Oklahoma: Organized, 1935. Headquarters, Oklahoma City, Okla. Meets, Second Monday. President, W. W. Frankfurt, 519 Apco Tower, Oklahoma City. Secretary, J. H. Spaan, Jr., 1303 N. Klein, Oklahoma City 6.

Ontario: Organized, 1922. Headquarters, Toronto, Ont., Canada. Meets, First Monday. President, E. G. Spall, 73 Simcoe St., Toronto 1. Secretary, H. R. Roth, 57 Bloor St., W., Toronto, Ont.

Oregon: Organized, 1939. Headquarters, Portland, Ore. Meets, Thursday after First Tuesday. President, E. R. Lokey, 1220 N. W. Glisan St., Portland 9. Secretary, A. N. Hoss, 621 S. W. Alder, Portland.

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Pacific Northwest: Organized, 1928. Headquarters, Seattle, Wash. Meets, Second Tuesday. President, L. L. Bysom, 1214 Eighth St., Bremerton. Secretary, D. O. Mead, 816 Arctic Bldg., Seattle 4.

Philadelphia: Organized, 1916. Headquarters, Philadelphia, Pa. Meets, Second Thursday. President, J. O. Kirkbride, Fourth & Locust Sts., Philadelphia 6. Secretary, M. E. Barnard, 12 S. 12th St., Philadelphia 7.

Pittsburgh: Organized, 1919. Headquarters, Pittsburgh, Pa. Meets, Second Monday. President, H. E. Park, 201 E. Carson St., Pittsburgh 19, Secretary, E. H. Riesmeyer, Jr., 231-33 Water St., Pittsburgh 22.

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Rocky Mountain: Organized, 1944. Headquarters, Denver, Colo. Meets, First Wednesday. President, Fred Janssen, 1123 W. Third Ave., Denver 9. Secretary, F. C. Allen, 4467 King St., Denver 11.

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St. Louis: Organized, 1918. Headquarters, St. Louis, Mo. Meets, First Tuesday. President, B. L. Evans, 571 Stratford Ave., University City. Secretary, C. H. Burnap, 1635 Syndicate Trust Bldg., St. Louis 1.

Shreveport: Organized 1948. Headquarters, Shreveport, La. President, W. E. FitzGerald, 939 Louisiana Ave., Shreveport. Secretary O. J. Dykes, Jr., 1012 Market St., Shreveport.

South Texas: Organized, 1938. Headquarters, Houston, Tex. Meets, Third Friday. President, D. M. Mills, Gray and Crawford, Houston 3. Secretary, L. L. Ladewig, Box 1188, Houston 1.

Southern California: Organized, 1930. Headquarters, Los Angeles, Calif. Meets, Third Wednesday. President, R. A. Lowe, 6381 Hollywood Blvd., Los Angeles 28. Secretary, John L. Blake, 1700 S. Bedford St., Los Angeles 35.

Southwest Texas: Organized, 1946. Headquarters, San Antonio, Tex. President, F. C. Benham, Jr., 702 S. Flores St., San Antonio. Secretary, L. S. Pawkett, 810 Insurance Bldg., San Antonio 5.

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Utah: Organized, 1944. Headquarters, Salt Lake City, Utah. Meets, First Wednesday. President, E. V. Gritton, 2470 S. 15th, Salt Lake City 5. Secretary, C. E. Ferguson, 838 Garfield Ave., Salt Lake City.

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Virginia: Organized, 1946. Headquarters, Norfolk, Va. Acting President, J. E. White, 7607 Gloucester Ave., Norfolk 5. Secretary, D. E. Phillips, P. O. Box 674, Newport News.

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Washington, D. C.: Organized, 1935. Headquarters, Washington, D. C. Meets, Second Wednesday. Washington, D. C. President, A. S. Gates, Jr., 621 N. Jackson St., Arlington, Va. Secretary, J. G. Muirheid, 2013 New Hampshire Ave., N. W., Washington 9.

Western Michigan: Organized, 1931. Headquarters, Grand Rapids, Mich. Meets, Second Monday. President, H. W. Wolters, 820 Staples Ave., N.W., Kalamazoo 54. Secretary, C. V. Saunders, P. O. Box 671, Kalamazoo 99.

Western New York: Organized, 1919. Headquarters, Buffalo, N. Y. Meets, Second Monday. President, G. E. Adema, 39 W. Balcom St., Buffalo 9. Secretary, J. H. Bryce, 88 E. Girard Blvd., Kenmore.

Wisconsin: Organized, 1922. Headquarters, Milwaukee, Wis. Meets, Third Monday. President, W. A. Ouweneel, 514 E. Ogden Ave., Milwaukee 2. Secretary, H. F. Brinen, 2226 Kinzie Ave., Racine.

Student Branches

North Carolina State College: Organized 1948. Headquarters, Raleigh, N. C. President, H. W. Powell, Jr., 144 Trailwood, Raleigh. Secretary, C. E. Jones, Box 3221, North Carolina State College, Raleigh.

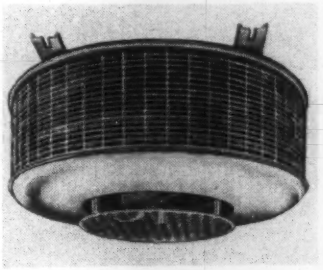
Purdue University: Organized 1948, Headquarters, W. Lafayette, Ind. President, R. G. James, 411 E. 37th St., Indianapolis. Secretary, H. H. Clark, Jr., 519 State St., W. Lafayette.

Texas A. & M. College: Organized, 1946. Headquarters, College Station, Tex. Meets, Third Thursday. President, S. E. Ammons, 212 13th Ave., No., Texas City. Secretary, C. P. Howard, Box 1706, College Station.

EQUIPMENT DEVELOPMENTS

For your convenience in obtaining more information about any of this equipment, see coupon on this page. Add the new products and companies listed here to your Directory Section which you received in your January 1948 Heating, Piping & Air Conditioning and thus keep your records of sources of supply up to date throughout the year. Single asterisk indicates equipment not listed in Directory Section; double asterisk equipment and manufacturer not listed.

Circular Design Used in Comfort Cooler



No. 3769—The new model 100 CC comfort cooler, developed by Merchant & Evans Co., 2035 Washington Ave., Philadelphia 46, Pa., can be mounted singly or in multiples on a ceiling, and is designed to circulate cooled, dehumidified air in a 360 deg radius along the ceiling and down to the floor. Refrigerant and condensate lines are connected near the top of the unit and a built-in pump is said to assure positive discharge of the condensate.

The unit is designed to handle 900 cfm and is rated at 1.10 tons, for 50 deg refrigerant temperature, and 2.04 tons for 35 deg refrigerant temperature. It is powered by a 1/15 hp motor, is 28 in. in diameter, and is 15⅝ in. high.

Develops New Equipment for Inert Gas-Shielded Arc Welding

No. 3770—Air Reduction Sales Co., 60 E. 42nd St., New York 17, N. Y., has developed a new water cooled "Heliweld" manual electrode holder, a machine type electrode holder, and an automatic unit.

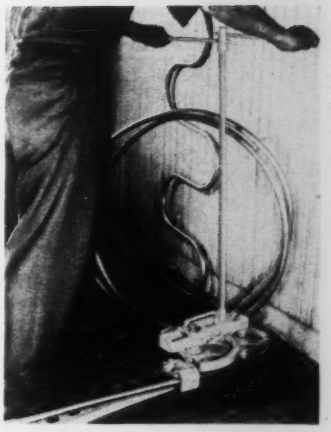
The manual holder, designed for continuous production work with the inert gas-shielded arc welding method, features water cooling, an all-plastic exterior, a gas cap made of nonconducting material, and a flexible cable encased in a translucent plastic cover. According to the company, cooling of the holder is highly efficient and extends almost to the point at which the electrode is gripped. It handles tungsten or carbon electrodes up to 3/16 in. diameter.

The new machine holder, which is also designed for welding light metals such as stainless steel, aluminum, and magnesium, is mounted on a radiagraph or other suitable travel mechanism for moving along the joint or for fixed position use with the work moving beneath the arc.

The automatic unit, designed for continuous operation, consists of a holder, holder carriage, and a control box. It is fully automatic and electronically controlled. The outstanding feature, says the manufacturer, is that it automatically maintains constant arc length.

These units have a welding current capacity of 300 amp. Both the manual and machine holder may be used with a-c or d-c welding current while the automatic unit, because of its electronic circuits, operates on d-c only.

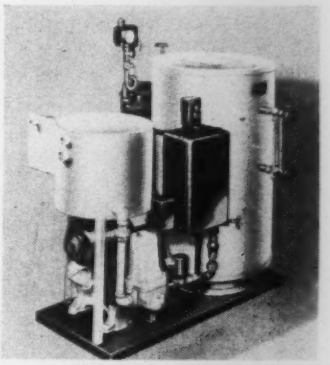
Tube Bender Designed for Radiant Heating



No. 3771—A lightweight portable copper tube bender is now being produced by Tal Bender, Inc., 417 N. Water St., Milwaukee 2, Wis.

Designed for radiant heating installations, the manufacturer states that it will bend both hard and soft copper tubing (types *K* and *L*) to a 180 deg return bend. Available in various radii for ⅜ to 1 in. sizes, the unit enables the operator to work with a continuous coil almost flush to the wall and to drop the finished bend out of the machine at the desired location. On the job changeover of formers and rollers from one size to another is said to be a simple matter.

Develops Electric Steam Generator for Commercial, Industrial Applications



No. 3772**—Custom Engineering Co., 3138 W. Slauson Ave., Los Angeles 43, Calif., has developed a line of electric steam generators in sizes from ½ to 25 hp. According to the company, the electrical features comply with the National Electric Code and the boiler is designed in accordance with the Power Boiler Code of the American Society of Mechanical Engineers. It is further stated that the only heat loss from the generator is by radiation, the shell being thoroughly insulated, so that "the thermal efficiency of the unit is 98 percent."

Heating, Piping & Air Conditioning [6-48]
6 N. Michigan Ave., Chicago 2, Ill.

Please ask the manufacturers to send me more information about the equipment mentioned under the following reference numbers in Equipment Developments and Recent Trade Literature

(Circle each number in which you are interested)

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3789	3790	3791	3792	3793					
6855	6856	6857	6858	6859	6860	6861	6862	6863	6864
6865	6866	6867	6868	6869	6870	6871	6872	6873	6874
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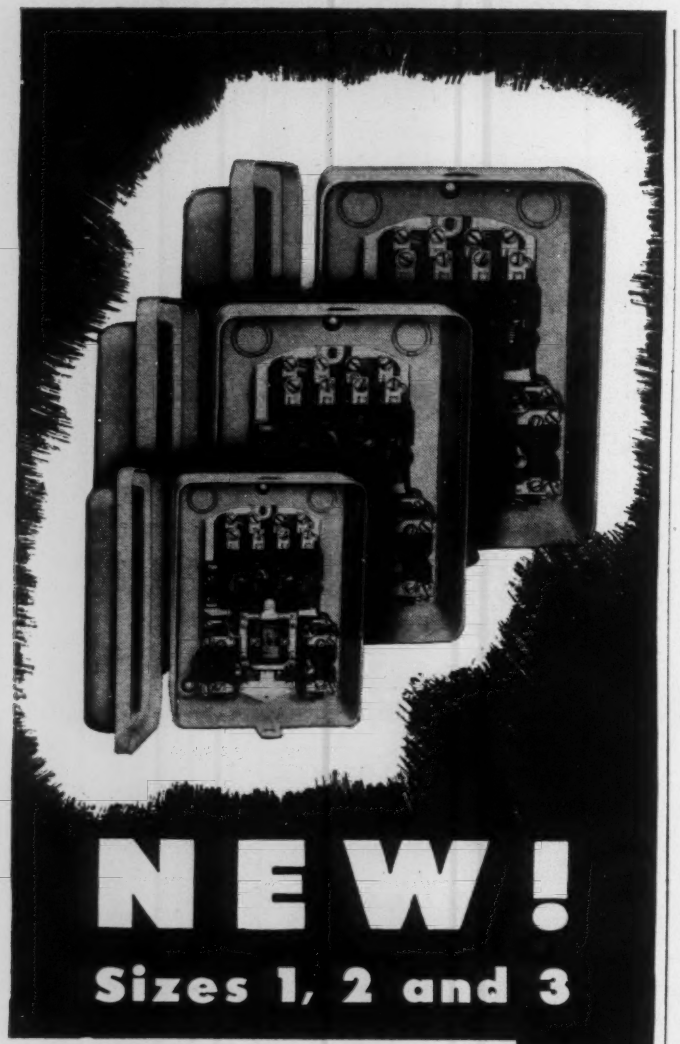

NEW!

Sizes 1, 2 and 3

Ward Leonard a-c Magnetic Starter Is Small in Size, Big in Performance

For fast, positive, dependable across-the-line starting of non-reversing motors, install Ward Leonard 4110 a-c Magnetic Starters.

Unit construction. Open or enclosed type. Accessible components. Double-break silver-to-silver main contacts. Thermal relays give overload protection. Local or remote control.

Ward Leonard Starters are "Result-Engineered." By modifying a basic design, Ward Leonard can give you the results of a special . . . for the cost of a standard.

Write for Bulletin 4110. Ward Leonard Electric Co., 24 South St., Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

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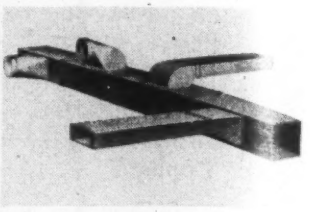
RESISTORS · RHEOSTATS · RELAYS · CONTROL DEVICES

This fully automatic unit is said to be especially suitable for use in dairies, bottling plants, garment industries, hospitals, laboratories, restaurants, laundries, etc.

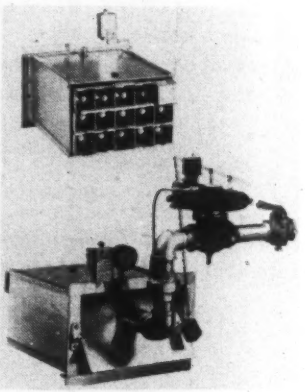
Offers Prefabricated Ducts and Fittings for Forced Air Heating



No. 3773—The new "Lamneck Simplenic" system of prefabricated ducts and fittings for forced air heating systems is now available from Clayton & Lambert Mfg. Co., 1701-1715 Dixie Highway, Louisville 10, Ky

Inasmuch as the system includes a complete line of standardized fittings, it is said to greatly simplify stocking, both as far as space requirements and investment of money are concerned. In addition, the ease of sizing, assembly, and erection is said to constitute an outstanding advantage. According to the manufacturer, the system was developed by the National Warm Air Heating and Air Conditioning Association.

Stainless Steel Mixing Plate Used in Gas Burner



No. 3774**—Five sizes in the new series SP gas burners offered by B. P. Lientz Co., P. O. Box 2275, Kansas City, Mo., are designed to provide a range from 150,000 to 2,000,000 Btu per hr for steam and hot water boilers, warm air furnaces, and special heat applications. The new units, designed for conversion and commercial installations, are identical in basic design to the company's large industrial burners. The use of stainless steel in the mixing plate is said to eliminate warping and cracking of the plate. Multiple orifice burner tips, also of stainless steel, are designed to permit a more intimate contact of gas and air.

Designed for natural gas, manufactured gas, or liquefied petroleum, the burners are furnished as a packaged unit and include a flame safety system, a built-in pilot, and the necessary controls for fully automatic operation.

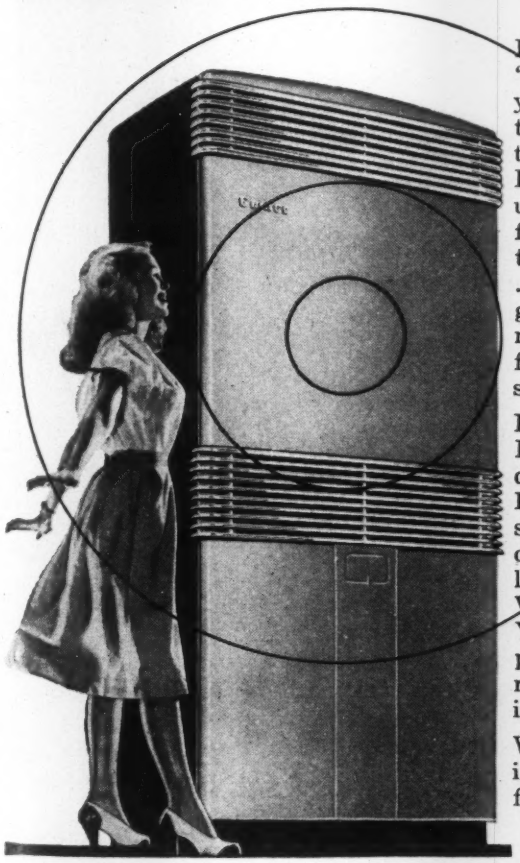
Protective Coating Air Dries Without Baking

No. 3775—Two new types of anti-corrosive "Zincilate" coatings, developed to air dry without baking, have been announced by Industrial Metal Protectives, Inc., Dayton 2, Ohio.

Accelerated laboratory tests are said to indicate that the coatings will give 20 year protection against corrosion and that they are sufficiently flexible so that sheets, pipes, and forms can be bent double without breaking the protective coating. Final coatings of enamel, paint, or wrinkle finishes may be applied over the product after 5 min. of air drying, and both coatings may be baked at the same time. It is adaptable to production line application by dipping, spraying, brushing, or roller coating, with conventional finishing equipment.

In these leading business magazines

7½ MILLION OF THE BEST "PROSPECTS" FOR AIR CONDITIONING in America are Reading why GEMCO'S THE BUY



If you're selling Gemco "packaged" Air Conditioners, you've felt the effect of national advertising directed to the *people who sign the orders.* It's a direct story of the unique, years-ahead Gemco features that make air conditioning practical to *buy now* . . . Your best prospects are getting this message, in the magazines shown at right. A few desirable territories are still open.

Remember, too—Gemco Dealers automatically become dealers for MIRACULA, the Heat Pump that caused a sensation at the recent Air Conditioning Exposition in Cleveland and the Heating and Ventilating Exposition in New York. Dealers are now making pilot installations, so as to be ready for the volume market in 1949.

Wire or write for complete information on the Gemco franchise *now.*

Gemco

AIR CONDITIONING

GENERAL ENGINEERING & MANUFACTURING CO.

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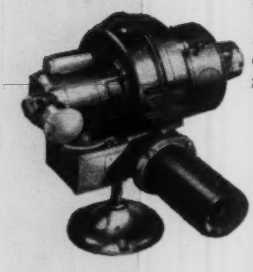
ALDRICH HEAT-PAK PRODUCTS

ALDRICH

OIL BURNERS

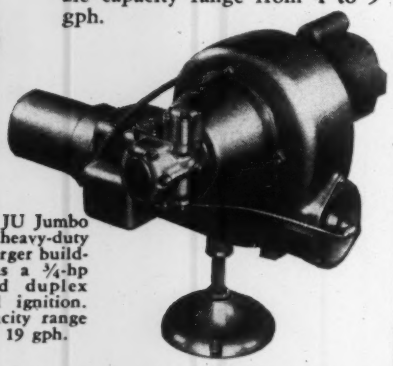
MODELS AX-1 AX-2 AX-3

The AX Models cover 90% of all conversion needs in residences and many commercial and industrial jobs. Low-cost interchangeable draft tube assemblies and air cones enable quick modification covering 3 nozzle capacity ranges: AX-1, .75 to 1.35 gph; AX-2, 1.35 to 2 gph; and AX-3, 2 to 5 gph. Various lengths of draft tubes available. You can meet a wide range of customer's requirements with a minimum stock.



The Model BX Burner is a larger size than the AX, covering a nozzle capacity range from 4 to 9 gph.

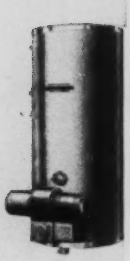
MODEL BX



MODEL JU

The Model JU Jumbo Burner is a heavy-duty model for larger buildings. It has a ¾-hp motor, and duplex nozzles and ignition. Nozzle capacity range is from 5 to 19 gph.

All Aldrich Burners are shipped complete with automatic controls, as specified, for warm air, steam, or hot water heating plants. Aldrich Products have gained wide and favorable acceptance, largely due to high quality construction that insures peak performance through a long and useful life.



BOILER-BURNER UNITS

Made in 6 sizes for hot water heating, steam heating, or hot water supply. Capacities 85,000 to 808,500 BTU per hour. Extremely well-made throughout.

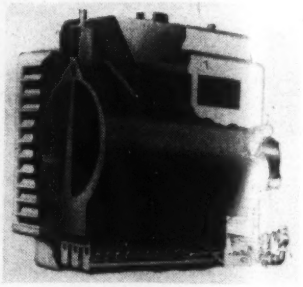
Write Today for Complete Data and Price Lists

ALDRICH COMPANY

OIL BURNERS • BOILER-BURNERS FOR HEATING AND DOMESTIC WATER SUPPLY

105 EAST WILLIAMS ST., WYOMING, ILL.

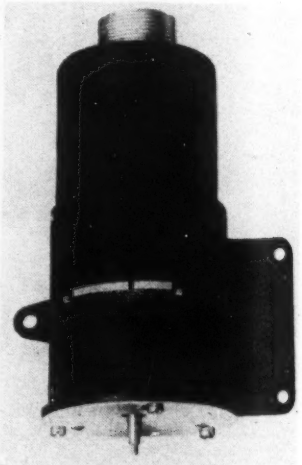
New Type Heat Exchanger Used in Gas Unit Heater



No. 3776—The new No. 120 "Humphrey" gas fired unit heater, rated at 120,000 Btu per hr input, has recently been announced by General Gas Light Co., Kalamazoo, Mich.

The chief feature of the heater, is the combustion chamber and heat exchanging unit which is designed to provide large surface area without the use of baffles and to offer little resistance to air flow through the heater. Other features include the use of a steel burner with "dual flame" stainless steel burner tips.

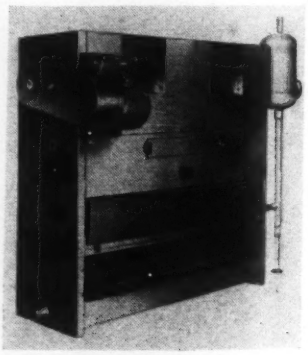
Remote Torque Unit Controls Industrial Machines, Processes



No. 3777—Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N. J., has announced a remote torque control system, that is adaptable to many industrial applications.

Already used in aircraft automatic pilot and engine control systems, boat steering apparatus, and humidity and liquid level controls, the system is designed to provide an electrical means for either manually or automatically transforming a very weak motivating force into control of a predetermined operation at a remote location with a high degree of accuracy. It is said to be suitable for controlling valves, gates, power sources, and signals.

Unit Cooler Line Designed for Many Industrial Uses



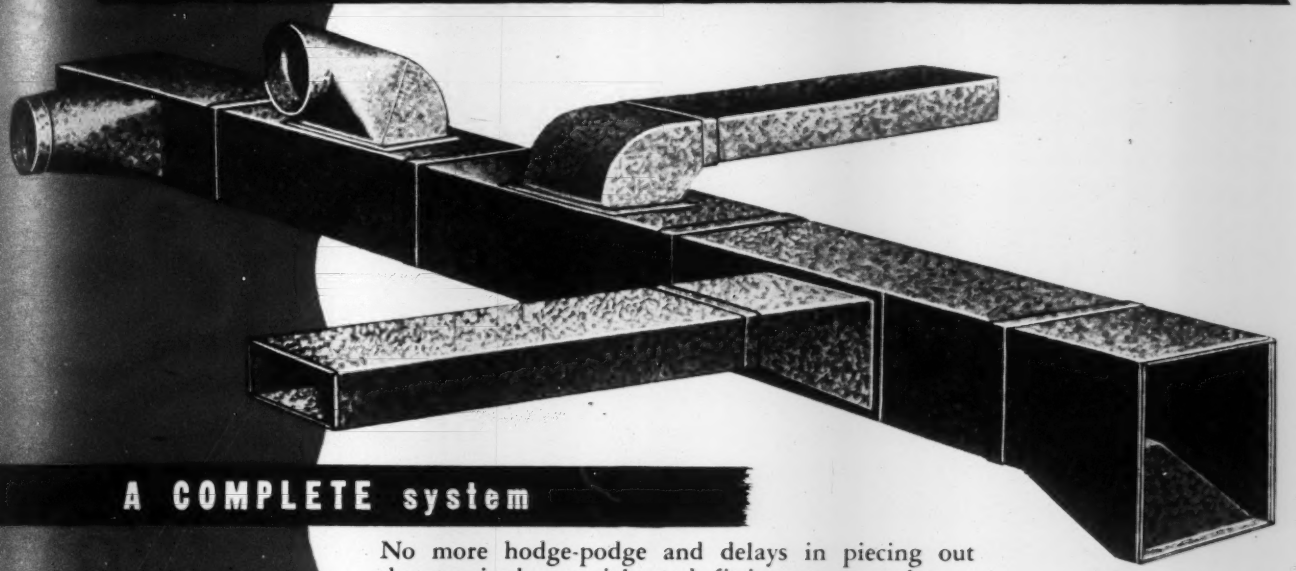
No. 3778—A new line of floor type industrial unit coolers featuring flexibility of installation has been announced by Acme Industries, Inc., Jackson, Mich.

The series B unit is offered for medium temperature applications where automatic defrosting is obtainable during the off period of the compressor. Series C units are recommended for use with lower temperature refrigerants and where defrosting may be accomplished by hot gas, water, or other liquids. Other models available include the continuous brine spray type for low temperature applications and prime surface units.

All unit coolers may be used with the direct expansion

You'll PROFIT better and WIN more sales with

The New Lamneck Simplenic System



A COMPLETE system

No more hodge-podge and delays in piecing out the required materials and fittings to complete a job, with a complete precision-matched Simplenic bill-of-materials!

SIMPLIFIED for efficiency

Only four duct fittings are required for serving every take-off application — and *no* increasers or reducers are needed! Fewer pieces to handle, fewer to assemble, no odd item stock problem!

FAST to install . . . .

Precision fabrication of all parts gives fast, accurate fitting — improved type, self-locking joint connections speed assembly operations — simple for even new men to install properly.

and LOW COST for volume sale!

Our production-line volume and material-saving design and your simplified job-figuring, material stocking, and installation ease, let you *buy* and *sell* the Simplenic System at best competitive prices!

WRITE TODAY for *Simplenic* descriptive price list and name of your nearest jobber.

ANOTHER GREAT C&L PRODUCT

CLAYTON & LAMBERT MFG. CO.
1762 DIXIE HIGHWAY • LOUISVILLE 10, KY.

Lamneck TRADE MARK

FURNACE PIPE AND FITTINGS

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Save Oil

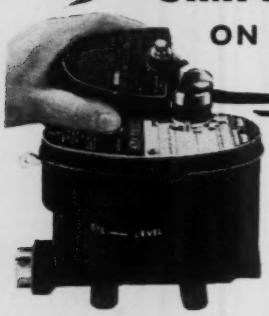
FOR YOUR CUSTOMERS with this popular-selling

THERMOSTATIC HEAT REGULATOR SET

Now, while your heater customers are totaling last season's heating costs, is the time to suggest installing an A-P THERMOSTATIC HEAT REGULATOR Set. You can promise substantial savings, plus the added comfort of steadier room temperatures, with this set on any vaporizing burner oil heater using A-P Safety Oil Controls. This includes new heaters as well as those made since 1939 — using A-P Controls No. 240-Y, U, or D series.

Here's a rich accessory sales opportunity — with a prospect list you've been building up for EIGHT YEARS . . . Heater owners waiting and watching for AUTOMATIC Temperature Control to truly modernize their vaporizing burning oil heaters.

SIMPLE TO INSTALL ON NEW OR OLD HEATERS



Just mount Electric Conversion Top on present A-P Manual Control. Then connect to modern Wall Thermostat and Transformer. "Sales Package" contains all accessories.

WRITE for complete Selling Material on A-P Oil Control Accessories — including A-P Thermostatic Heat Regulator Sets, Oilifter Automatic Fuel Handler, and Fuel Oil Trap-It.

AUTOMATIC PRODUCTS COMPANY

2460 North Thirty-Second Street, Milwaukee 10, Wisconsin

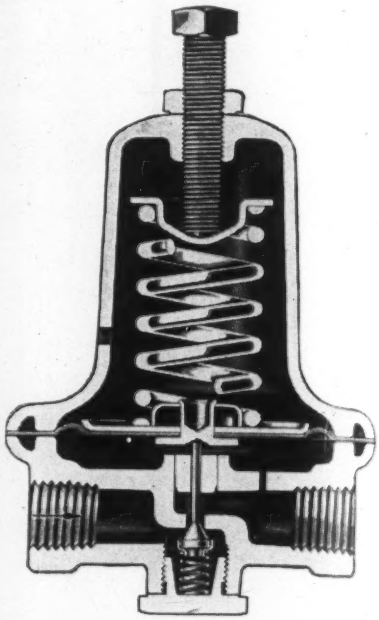
040 C

DEPENDABLE OIL CONTROLS

DESIGNED TO ELIMINATE SERVICING

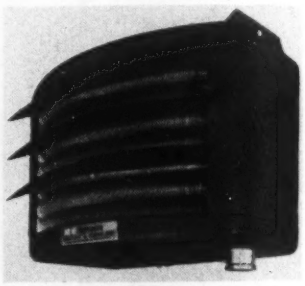
of "Freon-12" or with flooded ammonia operation, except for the prime surface units which are specifically designed for use with ammonia.

Small Volume Reducing Valve is Self-Contained



No. 3779 — The type 67 reducing valve for air or gas service applications is now available from Fisher Governor Co., Marshalltown, Iowa. It is designed to provide superior close pressure maintenance under exacting conditions on applications such as instrument air supply service and panel pneumatic loading for remote hand control. Available in the ¼ in. size only, its capacity is 6 cfm at 100 psi inlet, 5 psi reduced pressure. It is designed for initial pressures up to 250 psi at 150 F and reduced pressure ranges are from 3 to 20, 15 to 40, and 25 to 75 psi.

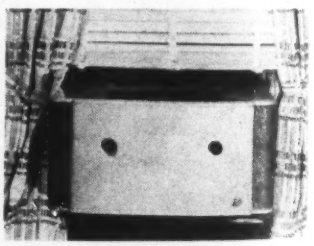
Diffusion Type Unit Heater Features Curved Face



No. 3780—The diffusing type unit heater offered by Merchant & Evans Co., 2035 Washington Ave., Philadelphia 46, Pa., is said to provide wide angle diffusion by the use of a 60 deg rounded front face.

Designed for use with low pressure steam or hot water, the heater is available in three models; 6 WH, 12 WH, and 34 WH, having capacities of 35,280, 75,000, and 181,000 Btu per hr respectively. Heat transfer is accomplished by the use of copper finned tubing.

Sealed Radial Compressor Used in Room Conditioner



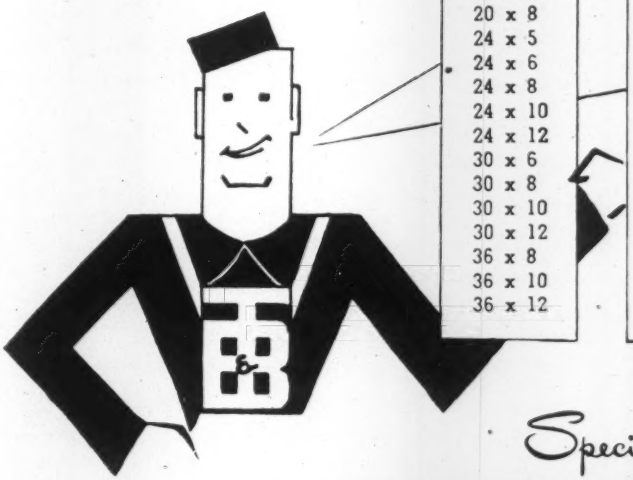
No. 3781—A new air cooled room air conditioner is now in production at Airtemp Div. of Chrysler Corp., 1119 Leo St., Dayton 1, Ohio. This new ¾ hp unit is said to be ideal for offices and to be suitable for either window installation or for location inside a room. The unit features the company's sealed radial compressor on special mountings designed for quiet operation.

Tri-Flex
26 sizes

8 x 4
10 x 4
10 x 6
12 x 4
12 x 5
12 x 6
14 x 4
14 x 5
14 x 6
16 x 5
16 x 6
20 x 5
20 x 6
20 x 8
24 x 5
24 x 6
24 x 8
24 x 10
24 x 12
30 x 6
30 x 8
30 x 10
30 x 12
36 x 8
36 x 10
36 x 12

Aerovane
20 sizes

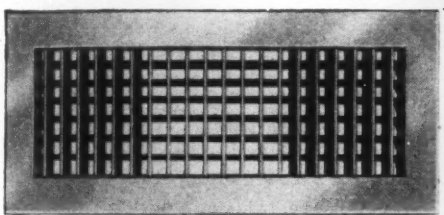
10 x 6
10 x 8
12 x 6
12 x 8
12 x 12
18 x 6
18 x 12
18 x 18
24 x 12
24 x 18
24 x 24
30 x 12
30 x 18
30 x 24
36 x 18
36 x 24
36 x 30
48 x 24
48 x 30
48 x 36



TAKE ADVANTAGE OF COST SAVINGS WHEN YOU SELECT AIR CONDITIONING GRILLES and REGISTERS

Specify... Install

T&B STANDARDS



FOR SUPPLY

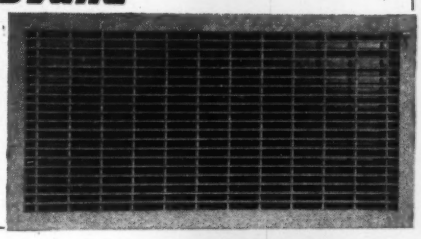
Available as Grille, Double Deflection Grille (shown), Multi-Shutter Register, or Double Deflection Multi-Shutter Register

Tri-Flex

Aerovane

FOR RETURN

Available as Grille, or Register (shown)



When you plan the grilles and registers for your next air conditioning job, plan efficiency with popular T&B STANDARDS . . . your answer for practically all commercial, industrial and institutional requirements.

Standardization of TRI-FLEX and AEROVANE sizes, resulting in mass production methods, means real savings . . . an important consideration in the face of rising construction costs.

And there are other advantages. On-the-shelf stock assures quick delivery . . . standardization eliminates guesswork in selecting correct sizes.

For detailed description, engineering data and complete information that will aid specifying and installing, write for a copy of Bulletin 47TF.

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Prompt Service
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Norman J. Learned, President
LeValley McLeod, Inc.

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THE Herman Nelson Corporation and carefully selected distributors like the LeValley McLeod, Inc., of Elmira, New York, do more than provide you with quality heating and ventilating products. Their friendly, close working partnership means that you get prompt delivery of quality Herman Nelson products and carefully planned merchandising cooperation which has increased profits for dealers and contractors all over America.

Effective advertising, literature and sales promotion campaigns create an acceptance for Herman Nelson products that leads to increased sales and profits. Furthermore, you are assured of immediate service from the sales and engineering departments of your nearest Herman Nelson distributor. Here's another advantage. Your Herman Nelson distributor will supply you with both equipment and installation materials promptly from stock.

Contact a Herman Nelson distributor and profit from his complete cooperation. You'll find his services increase your profits through more sales of heating and ventilating equipment.

THE HERMAN NELSON CORPORATION
Since 1906 Manufacturers of Quality Heating and Ventilating Products
MOLINE, ILLINOIS

Clock Thermostat Plugs Into Nearest 110 Volt Outlet

No. 3782 — A "plug-in" clock thermostat that can be installed in less than 5 min has been announced by Minneapolis-Honeywell Regulator Co., 2504 Fourth Ave., S., Minneapolis 8, Minn.

Containing an electric clock for automatically lowering room temperature at night and raising it again in the morning, the thermostat is a modification of the company's "Chronotherm," but uses a 110 volt clock motor. The unit is equipped with a universal wall plate designed to accommodate electrical connections with existing thermostat wires. An electrical cord is furnished for plugging into the nearest electrical outlet.

According to the company, the universal design of wall plate makes the installation of the new thermostat applicable to almost all types of existing thermostat mountings.

Check Valve Prevents Refrigerant Backflow in Two-Temperature Systems

No. 3783—A new refrigerant check valve, model 234, is offered by Automatic Products Co., Milwaukee 10, Wis., for installation in the suction line of the low temperature evaporator of a two temperature system. It is designed to prevent backflow of the refrigerant from the high to low temperature evaporator during the off cycle period.

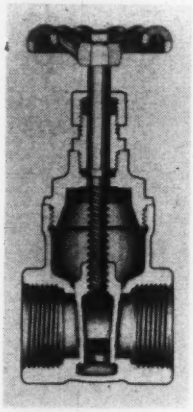
Pressure drop through the check valve is said to be negligible. Positive seal-off at all pressures, says the company, is accomplished by combining a metal valve disc with a carefully finished brass seat. Because the disc and seat resist corrosion, the unit can be used with all refrigerants.

Introduces New Air Conditioning Compressor

No. 3784—Shown here is the amount of ice which would be needed everyday to produce the same amount of cooling as the new 2 hp air conditioning compressor introduced by the Sturtevant Div., Westinghouse Electric Corp., Hyde Park, Boston, Mass.

The 300 lb compressor unit and the 2 tons of ice are being inspected by some of the distributors who attended the company's first postwar air conditioning conference.

Nonrising Stem Construction Used in Brass Gate Valves



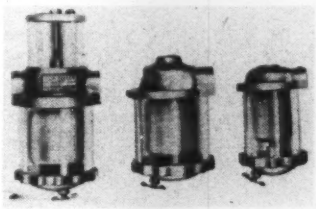
No. 3785—Crane Co., 836 S. Michigan Ave., Chicago, is now marketing a new line of 100 lb brass gate valves which are furnished in two types, the No. 410 screwed end valve (in sizes ¼ to 2 in.) and the No. 1320 solder joint valve in sizes from ⅜ to 2 in. These are nonrising stem valves and are intended to replace the company's No. 449½ and 1319 valves which are discontinued.

The body design of the new valves is said to be an innovation, incorporating features rarely found in low pressure valves. The upper portion of the body is cylindrical in shape, similar to the design that has been used in steel valves for the highest pressures. The cylindrical shape of the body is also used to reinforce the seats against the wedging action of the disc. This, says the company, safeguards against distortion of the body when the valve is being seated.

The ends of the solder joint design are made to fit closely over hard copper tubing so that solder will flow into the joint by capillary action, the solder being fed into the joint from the outer end of the socket.

The screwed valves are intended for all types of use, such as water lines, steam and hot water heating lines, gas lines, and fuel oil lines. The solder joint valves are suitable for the same services, except that they are not recommended for gas or fuel oil because of the fire hazard in working with solder. Specific working recommendations allow the valves to be used for saturated steam at working pressures up to 100 psi and for water pressures up to 125 psi at 200 F.

Plastic Bowl Used in Air Line Filters, Lubricators



No. 3786**—A new, simplified line of transparent, plastic bowl filter-lubricators and single air line filters designed to increase the life and efficiency of air operated equipment of all types has been developed by CCA Products, Inc., P. O. Box 671, Glendale 5, Calif.

The new plastic bowl elements feature enlarged areas of both filter bowls and all wool felt filter gloves to increase operating capacity and reduce maintenance service as well. A single hold down screw in each case is used to simplify and speed up disassembly when it becomes necessary to service the equipment or change filter gloves. "O" rings are used to provide sealing of the component filter parts. The "Filter-Lube" unit is an air line filter and automatic lubricating injector combination, completely air actuated. A precision regulating or metering device is said to permit the correct amount of oil to mix with clean, dry, operating air.

Nonbreakable plastic bowls, light weight, compact installation, and protection against sludge, dust, sand, oil, rust, scale, and free moisture are claimed for the units. Four sizes are available: ¼, ⅜, ½ and ¾ in. IPS.

ADSCO
DUAL STEAM
TRAPS
• Valve and float mechanism mounted on cover for quick, easy cleaning. Stainless steel valve and seat reversible for double life. Inlet and outlet connections arranged for convenient piping. Complete line for wide range of capacities. For all steam pressures to 125 psi. Available in float only type or float and thermostatic type.
Write for Folder 247
AMERICAN DISTRICT STEAM CO.
NORTH TONAWANDA, N. Y.
SINCE 1877

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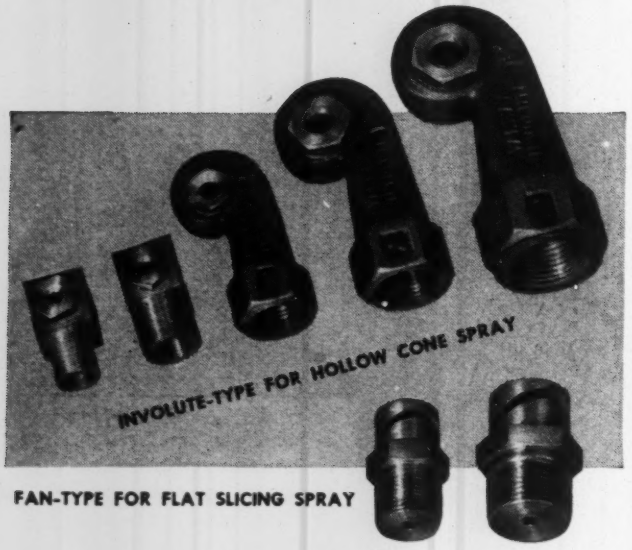
for • SPRAYING
• WASHING
• RINSING
• COOLING
• AIR CONDITIONING

YARWAY

SPRAY NOZZLES



FAN-TYPE FOR FLAT SLICING SPRAY

Get the most out of your Spraying Equipment with minimum power . . . with efficient spraying.

Use Yarway Nozzles. No internal vanes or other restrictions to clog or hinder flow. Two types—Yarway Involute-type producing a fine hollow spray with minimum energy loss, and Yarway Fan-type producing a flat fan-shaped spray with time-saving slicing action for cleaning.

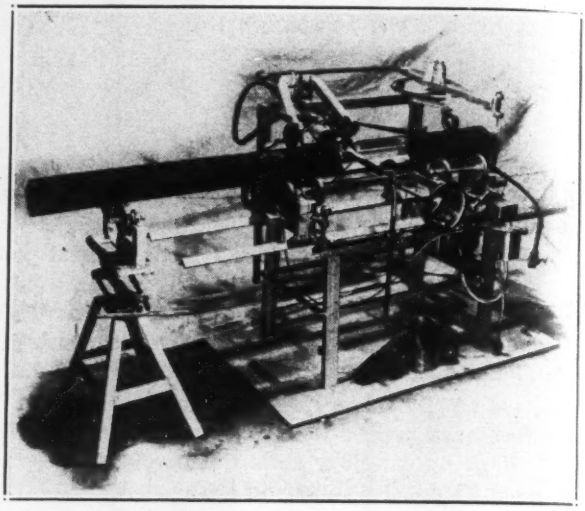
Wide range of standard sizes and capacities. Cast or machined from solid bar stock.

Thousands in use. Write for new Bulletin N-616.

YARNALL-WARING COMPANY
107 Mermaid Avenue Philadelphia 18, Pa.

Pipe Cutting Machine Utilizes Oxyacetylene Torch

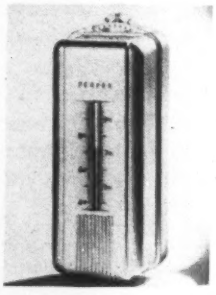
No. 3787**—The Stratford Pipe Scalloper Machine Co., 49 Oakland Pl., Stratford, Conn., has developed a machine for cutting pipe. According to the manufacturer it will cut, contour, and chamfer in one operation any steel pipe from 1 to 12 in. in diameter and from 6 in. to any desired length. Cutting is accomplished by means of a standard oxyacetylene torch. The rotational speed of



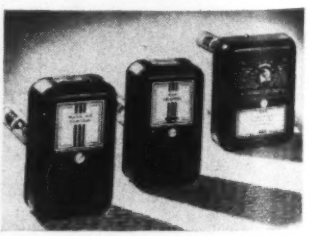
the pipe is related to the torch so as to produce a "ready-to-weld" surface. For contour cuts, the torch bearing pantograph follows a standard circular template. For square cuts the pantograph is locked at any desired position and chamfer control is automatic for all cuts. Powered by a ¼ hp single speed reversible motor, the machine is designed to cut 1.15 fpm of pipe, to require only negligible set up time, and to eliminate touch up time.

Develops New Thermostat and Warm Air Heating Controls

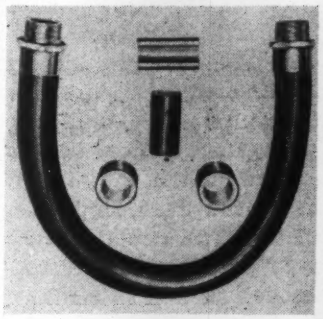
No. 3788—The Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis., has developed a new "Magic Dial" thermostat, series 165, designed to operate with the company's full line of primary controls on oil, gas, stoker or hand-fired coal heating installations. According to the company, both temperature and cycling adjustments are made on twin dials surmounting the die-cast thermostat case. A wide range of cycling adjustments from a short "on heat" period to a long "on heat" period can be made on the upper dial.



Also recently developed are the company's new warm air limit control, series 404; a new fan control, series 454; and a new combination fan and limit control, series 474. Each of these new controls is said to feature a simplified design and completely enclosed twin contact switches.



Offers Synthetic Resin Hose for Many Solvents, Gases



No. 3789—Resistoflex Corp., Belleville 9, N. J., has announced that its hose, made with a tube stock of compar, is now available in larger sizes for conducting solvents and gases. A variety of constructions are offered up to 1½ in. inside diameter for low pressure and medium pressure applications. Most of these constructions are produced with or without a static discharge wire.

The tube stock is said to be suitable for handling organic solvents of all kinds, including aromatic hydrocarbons such as benzol, toluol, and zylol; chlorinated solvents, including carbon tetrachloride, perchlorethylene, and trichlorethylene; the new insecticides, DDT and hexaethyltetraphosphate; and chemicals of the ester group, such as butyl acetate and propionate. In addition to being unaffected by fuels, oils, petroleum spirits, naphtha, ether, and carbon disulphite, the company states that some compar stocks are completely impermeable to many widely used gases such as "Freon 12," "Freon 22," and methyl chloride.

Compar is not the name of one material, but refers rather to a group of synthetic resins or elastomers, each of which can be compounded or modified to meet specific requirements.

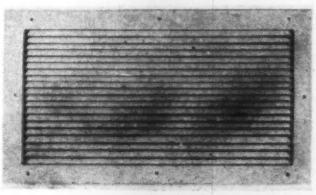
Power Drive Operates Hand Pipe Cutters, Threaders, Reamers

No. 3790—The Ridge Tool Co., Elyria, Ohio, has announced the development of a new power drive for operating hand pipe cutters, threaders, and reamers. Named the "Rigid Porta-Power Drive," it weighs 100 lb and, according to the manufacturer, is the lightest machine of its kind.

It has a three jaw, lathe type scroll chuck in front and a self centering workholder in the rear which turns with the pipe. Additional features include a special chuck wrench ejector, a bumpproof start and stop switch, and permanent sealed-in lubrication.

The power drive plugs into any 110 a-c light socket and its capacity is from ⅛ to 2 in. pipe and up to 8 in. pipe when used with the company's universal drive shaft.

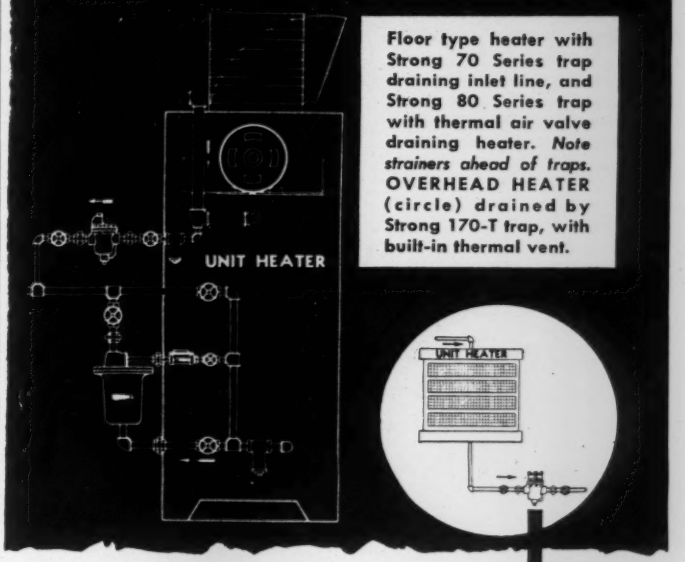
Ventilation Grille Restricts Vision



No. 3791—The "Louvaire" grille, designed primarily for installation in doors or partitions, is produced by the Stewart Mfg. Co., 610 Bloomfield Ave., Bloomfield, N. J.

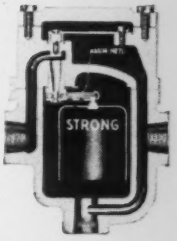
The horizontal, herringbone shaped bars of the grille are designed to permit passage of air for ventilation and at the same time to restrict vision. Grilles are made in all standard and extra large sizes and can be had in special color or plated finishes.

MORE HEAT · FASTER WITH LESS FUEL!



Floor type heater with Strong 70 Series trap draining inlet line, and Strong 80 Series trap with thermal air valve draining heater. *Note strainers ahead of traps.* OVERHEAD HEATER (circle) drained by Strong 170-T trap, with built-in thermal vent.

Accumulated condensate and trapped air can make unit heaters slow in starting. Uncertain drainage will seriously reduce heat output.



Strong 170-T trap with integral thermal air vent and new HI-CAP*Orifice. Available in sizes from ½" to 1¼".
*Reg. Trade Mark

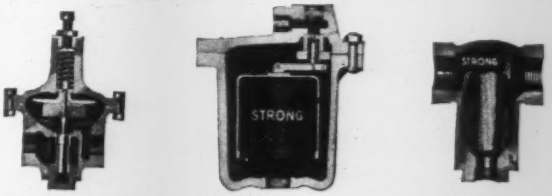
LEAKY TRAPS WASTE FUEL

While *all* Strong Inverted Bucket Traps will handle large amounts of air, the hookups shown here provide thermal air vents for practically instantaneous heating. Use of Strong traps, guaranteed for a year against leakage, insures positive drainage without waste of steam.

Completeness of the Strong line enables us to recommend exactly the right type and size of trap you need—open or inverted bucket, closed float, float-and-thermostatic (blast), etc.—semisteel, cast, forged and welded construction. Send your drainage problems to us for the *right* solution.

STRONG, CARLISLE & HAMMOND COMPANY
1392 West 3rd Street
Cleveland 13, Ohio
REG. TRADE MARK
ANUM-METL Reg. Trade Mark

OTHER STRONG STEAM SPECIALTIES



Reducing Valve | No. 80 Series Trap | Strainer

STRONG

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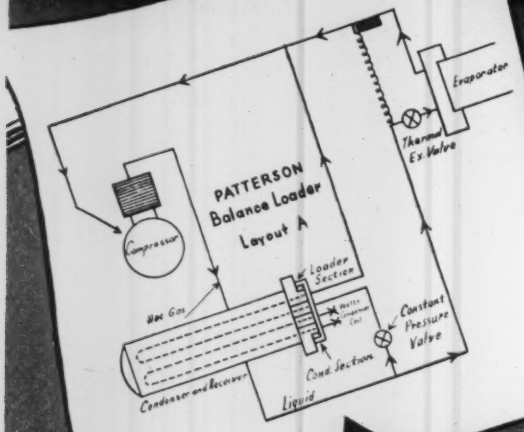
PATTERSON

BALANCE-LOADER



RECOMMENDED FOR BETTER CAPACITY CONTROL OF REFRIGERATING SYSTEMS

Featured by . . . Simplicity . . . Efficiency . . . and Complete Modulating Effectiveness from 0% to 100% full load. Adaptable to all variations in Refrigeration Systems.

In writing for further details, give information on capacity and evaporator temperature desired and a description of the system. The Patterson Balance-Loader can be installed in existing systems.

THE PATTERSON-KELLEY Company, Inc.

107 WARREN STREET, EAST STROUDSBURG, PA.

NEW YORK 17—101 Park Ave. PHILADELPHIA 3—1700 Walnut St.
CHICAGO 4—Railway Exch. Bldg. BOSTON 16—96-A Huntington Ave.

Representatives in Principal Cities

Coolant Pumps Developed for Machine Tool, General Use

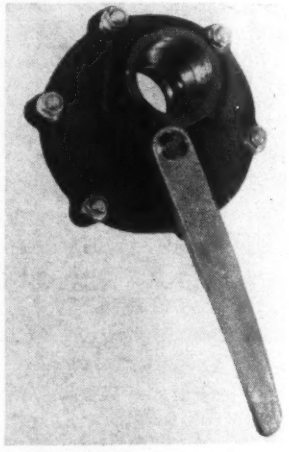
No. 3792—Two new seal-less type coolant pumps, a vertical submerged and a side wall mounted unit, have been added to the Norwood line of Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.

In general, these two new pumps, designed for general circulation duty and machine tool applications, are of the low head, single stage types close coupled to the motor. The double suction, open impeller, submerged type pump is constructed for vertical operation with flooded suction at all times. Known as the KF type, it is designed for use on machines requiring a large coolant tank. Said to be especially suitable for handling liquids containing considerable amounts of solids, chips, or suspended abrasive particles, this pump has no stuffing box.

The side wall mounted pump is designed for handling solids and gritty abrasive particles. It is commonly used on grinding and honing machines. This unit, the type KW, is available with two different length adapters depending upon the depth of the coolant reservoir.

Both pumps are available in ¼ to 1 hp sizes, 1 and 1¼ in., 10 to 70 ft heads, 1725 to 3450 rpm.

Shutoff Valve Designed for Quick Opening Operation



No. 3793**—Quick opening operation, leakproof characteristics, and heavy duty, all-steel construction are features of a new shutoff valve now being offered by Saval Co., 1915 E. 51st St., Los Angeles 11, Calif. Designed for butane and other similar service, the unit is also available for water or corrosive service with a heavy cast bronze body.

The valve is of 2 in. pipe size with a straight through opening of 1-11/16 in. diameter. An 18 lb force is required at the end of the handle to operate at 200 psi. A "Shear-Seal" principle is designed to provide a metal-to-metal seal that is self aligning and automatically to compensate for wear. According to the manufacturer, it is not critical to dirt because foreign materials are wiped away instead of lodging between the sealing surfaces.

The valve has only one basic moving part and requires no lubrication or packing adjustment.

Equipment Shorts

Wheatland Tube Co., Philadelphia, has reduced its price on buttweld pipe $2.00 per net ton, according to a recent announcement by E. P. Maneely, company president.

A new East Chicago, Ind. plant is now producing "Freon" refrigerants and officials of Kinetic Chemicals, Inc., state that it will increase production capacity by 60 percent. Shipments from East Chicago are in ton

capacity containers only. Smaller cylinders will continue to be shipped from the plant at Carney's Point, N. J. According to the company, raw materials are now in adequate supply for one year's production. The shortage of containers, however, is so acute that unless empties are returned, production and distribution face drastic cuts.

E. I. du Pont de Nemours and Co., Inc., has announced a $30,000,000 expansion of research facilities at its experimental station in Wilmington, Del. When the expansion is completed, the company expects to have 900 technical employees engaged exclusively in research work at the station. The entire job, including work on new and existing buildings as well as outside work, is to be completed in approximately two and one-half years.

More than 60 sales representatives recently attended a three day national sales conference held in Modine Mfg. Co.'s new administration building, Racine, Wis. The conference covered sales engineering, product analysis and application, advertising and market research. Another series of meetings was held for about 50 salesmen who were given training and background information on production, engineering, application, and the sale of company products.

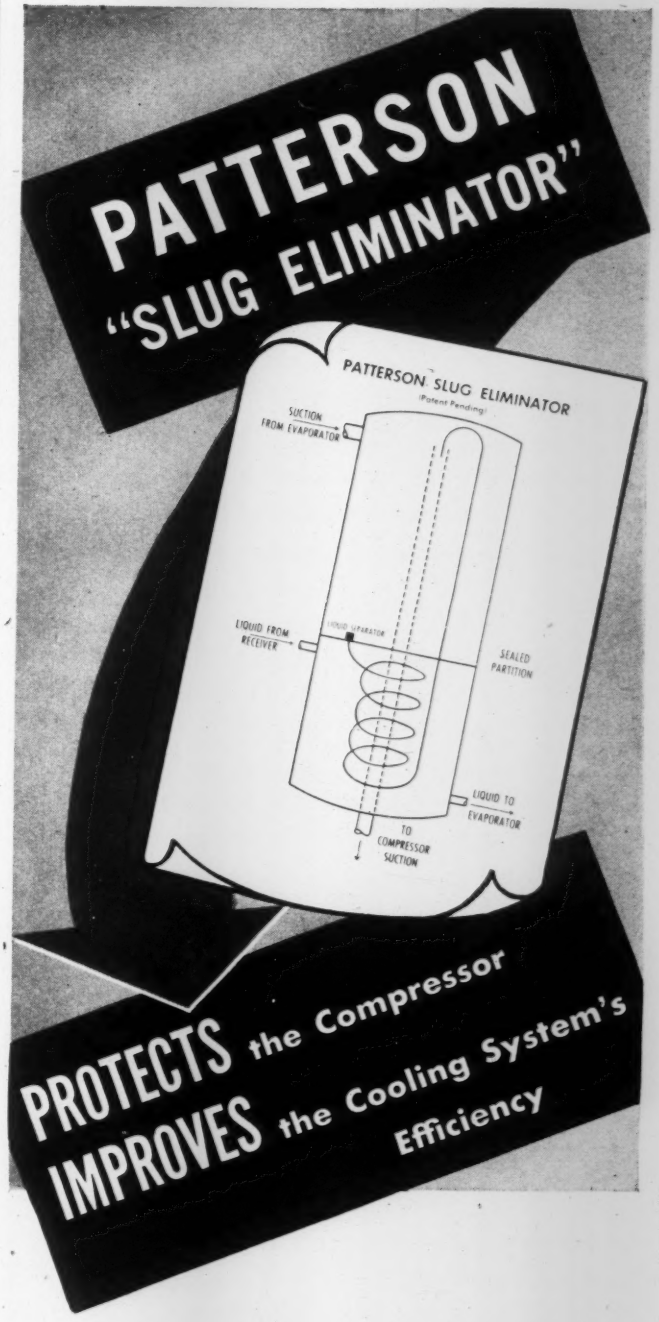
The name of the American Rolling Mill Co. was officially changed to Armco Steel Corp. at the recent annual meeting of stockholders. Since the organization has been a fully integrated steel company for many years, it is believed the new name will more fully reflect the exact nature and extent of the company's business. There will be no changes in the company's structure, personnel, policies, or activities.

Wolverine Tube Div. of Calumet & Hecla Consolidated Copper Co., Detroit, is constructing a new tube mill in Decatur, Ala.. Construction was begun last year and is expected to be completed this year. When finished, the mill will employ about 350 people and will be used to manufacture seamless nonferrous tube and to introduce the operation of a new electro-deposition unit for producing tube.

According to Reynolds Metals Co., Louisville, Ky., an intensive program to develop new uses for aluminum in industrial machinery is resulting in greatly increased interest in aluminum in this field.

Clayton & Lambert Mfg. Co., Louisville, Ky., has developed a hydraulic, multiple head drill which is used for drilling 148 holes simultaneously in the burners of its storage water heaters. According to the company, this has increased burner production 727 percent.

Wagner Electric Corp. has purchased a modern brick warehouse in St. Louis covering one entire city block. The building has four floors, a mezzanine, and a high-ceiling basement.



Order it for your cooler . . . existing or proposed.

The Patterson "Slug Eliminator" is positive in its liquid trapping action and by sub-cooling the refrigerant improves the efficiency of the system. It is simple and effective and has only one moving part — the separating valve.

In writing for further details, including prices and delivery, describe fully the cooler setup you wish to protect. We might have some installation suggestions.

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PIPE WRENCH TROUBLE AND EXPENSE

RIDGID pipe wrench housings are guaranteed forever against break or warp

● That guarantee means you can forget wrench housing bothers and repair costs when you buy RIDGIDs. Other RIDGID features make sense too—free-spinning adjusting nut; convenient pipe scale on hookjaw; instant-action non-slip, non-lock jaws and powerful comfort-grip I-beam handle. No wonder it's a favorite with millions of men all over the world who know fine tools. Sizes from 6" to 60"—try them and buy them at your Supply House.

RIDGID
WORK-SAVER PIPE TOOLS
THE RIDGE TOOL COMPANY • ELYRIA, OHIO

RIDGID *Strap Wrench won't mar polished pipe or tubing.*

EXECUTIVE CHANGES

The engineering consulting firm of Bueter & Wolff, Inc., 37 S. Wabash Ave., Chicago, has recently been formed by N. E. Bueter, mechanical engineer, and E. J. Wolff, electrical engineer. Mr. Bueter was formerly with Holabird & Root, Chicago architects, and more recently was director of sales for the Carrier Corp., Syracuse. He has a broad background of experience on many large buildings including offices, apartments, shops, schools, industrial plants, theaters, etc.

Mr. Wolff, who started in the electrical contracting field in the middle twenties, joined H. & R. in 1939, becoming chief electrical engineer for that firm in 1940. Between them they have been in responsible charge of electrical and mechanical design (including heating, ventilating, air conditioning, and fire protection) of many projects such as the Northwestern Technological Institute, Evanston, Ill., the Statler hotel, Washington, D. C., the U. S. Army facilities at Robbins Field, Ga., and the Standard Oil Co. research laboratories at Whiting, Ind.

Carrier Corp., Syracuse, N. Y., has formed Carrier Africa Ltd., a subsidiary company with headquarters in Johannesburg, Union of South Africa. Ralph S. Reynolds, associated with Carrier for 19 years, has been named general manager of the new firm. Ledyard Avery, who has represented the company in Singapore and Canada, is chief engineer.

Bell & Gossett Co., Morton Grove, Ill., has announced the appointment of William A. Boone as assistant sales manager. Mr. Boone has been with the company since 1940 and has been active in the industrial division. His headquarters will be at the home office.

Henry Valve Co., has appointed Evan Jones as chief engineer. Mr. Jones joined the company in 1945 and is a graduate of the Carnegie Institute of Technology.

The election of a representative group of technical employees of the organization as members of the firm has been announced by Albert Kahn Associated Architects and Engineers, Inc. G. H. Miehls, president, stated that the election effectuated a policy designed to permit assumption of increased responsibility in company operations by key employees.

J. W. Crossett has joined the development and research division of International Nickel Co., Inc. Mr. Crossett will follow railroad development work and succeeds the late F. P. Huston in that capacity. He is a member of the American Society for Metals and is a graduate of the University of Wisconsin, having received a bachelor's degree in chemical engineering in 1936.

The appointment of L. S. Maehling as district sales manager for western Pennsylvania, eastern Ohio, and

northwestern West Virginia has been announced by Richmond Radiator Co., New York City, an affiliate of Reynolds Metals Co. Mr. Maehling's headquarters will be in Pittsburgh.

J. A. Rodgers has been elected president of White-Rodgers Electric Co., St. Louis, manufacturer of electric controls. He succeeds L. F. Blough who is retiring as president to undertake other duties connected with the organization.

Raytheon Mfg. Co., Waltham, Mass., has announced the appointment of J. J. Tynan as sales manager of its commercial products division. K. V. Curtis has been named product manager and W. A. Gray continues in his present capacity as assistant sales manager.

N. S. Sanborn is now west coast district manager for York-Shipley, Inc., York, Pa. Mr. Sanborn, who served for over four years in the U. S. Army transportation corps as a major, has had wide experience in the heating field.

F. C. Boyce has been re-elected president of the D. J. Murray Mfg. Co., Wausau, Wis. Other officers elected at a recent meeting of directors are: M. P. McCullough, vice president; C. E. Staky, executive vice president and general manager; J. S. Alexander, treasurer; and A. W. Plier, secretary and assistant manager.

Alvin Dattle has been appointed regional manager in the east central states by Tenney Engineering, Inc., Newark, N. J. He will work with jobbers and manufacturers on equipment applications.

O. W. Bynum is now general sales manager of Carrier Corp., Syracuse. Since 1946 he has been manager of direct sales and will continue these duties in addition to directing the overall line organization of the sales division under A. P. Shanklin, vice president in charge of the sales division.

Alco Valve Co., St. Louis, has promoted R. B. McCrady, field engineer, from the company's Chicago office to take charge of its southwest territory which includes Texas, Oklahoma, and Louisiana. Mr. McCrady has spent his entire business career with the firm, except for three years in the U. S. engineer corps during the war.

Yarnall-Waring Co., Philadelphia, has named R. S. Pollard as sales manager of its steam trap division. The company has recently opened new sales offices in Cincinnati and St. Louis.

According to an announcement from the Wolverine Tube Div. of Calumet & Hecla Consolidated Copper Co., Inc., Detroit, L. H. Littlefield is now general sales manager. His headquarters is in the general offices in Detroit.

Speedy
PIPE CUTTING
IN TIGHT PLACES

RIDGID No. 42 has 4 cutting wheels for quarter turn cuts in cramped quarters

● When you run into those hard-to-get-at jobs, you'll like the fast easy action of the RIDGID No. 42. Its short handle makes it handy in cramped quarters — 4 wheel design allows fast burr-free cuts even where you have room for only a quarter turn. Special malleable housing won't distort; heavy-duty cutter wheels always track exactly. Ask for RIDGID No. 42 (1/2" to 2") or No. 44 (2 1/2" to 4") at your Supply House.

Regular heavy-duty RIDGID cutters in 5 sizes to 6" pipe.

RIDGID
WORK-SAVER PIPE TOOLS
THE RIDGE TOOL COMPANY • ELYRIA, OHIO

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BEND COLD PIPE FAST, RIGHT ON THE JOB

with a GREENLEE HYDRAULIC BENDER

Cut costs, speed piping installations this efficient, easy, economical way . . . make "on the job" bends in pipe up to 4½" with a GREENLEE Hydraulic Bender.

Simple to operate, easy to carry, set up, and shift from job to job . . . lets you bend pipe exactly where and when you want it. With a GREENLEE, *one man* makes smooth, accurate, uniform bends in a few minutes. You save greatly through elimination of many manufactured bends and fittings. And, in timesavings alone, the GREENLEE pays for itself quickly.

Ideal for radiant heating jobs, too. For complete details write for free folder S-121. Greenlee Tool Co., Division of Greenlee Bros. & Co., 2346 Twelfth Street, Rockford, Illinois.

TOOLS FOR CRAFTSMEN

GREENLEE

Executive Changes (Cont.)

H. S. Acton has been promoted to factory superintendent of Bogue Electric Mfg. Co., Paterson, N. J. He is now in charge of all manufacturing operations of the company.

The new sales manager of the Dri-Pipe Div., Mystic Adhesive Products, Chicago, is W. A. Givens. He is a graduate of the Lewis Institute and has been active in the air conditioning field for some time.

The Buildice Co. Inc., Chicago, has named R. B. Crawford as vice president and a member of the board of directors. He has been with the company as sales manager since last September and will continue in that capacity in addition to his new executive duties.

C. J. Reimer is now general purchasing agent for SKF Industries, Inc. He is a graduate of St. Joseph's College and is a member of the American Society for Metals.

M. M. Yarrington has been appointed general manager and F. C. Wittig assistant general manager of Aeroil Products Co., New York City. Both men are veterans of the company.

R. V. Baker, assistant secretary, has been named a vice president of Omega Machine Co., Div. of Builders Iron Foundry, Providence, R. I. He joined the company in 1935 as a development and experimental assistant.

The Lincoln Electric Co., Cleveland, has appointed G. F. Clipsham to the position of assistant to the president. He joined the company in 1929, was active as a welding engineer in various sections of this country, and in 1935 went to England to organize and head the Lincoln Electric Co., Welwyn Garden City, Herts, England.

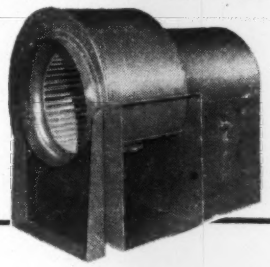
A series of promotions within the ranks of the technical sales personnel of Air Reduction Sales Co., New York City, has been effected. Dr. G. V. Slottman, formerly manager, technical sales division, is now assistant to the vice president. He has been with the company 12 years and holds degrees from Massachusetts Institute of Technology and the University of Berlin. He is a former professor of chemical engineering at MIT. S. D. Baumer, formerly assistant manager, technical sales division, is manager of that division, and E. H. Roper is assistant manager.

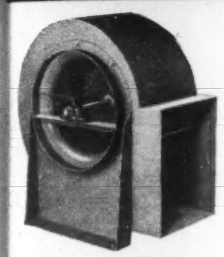
L. J. Mueller Furnace Co., Milwaukee, has named Ronald F. Marsden as director of purchases. Prior to joining the company he conducted his own business of procuring and subcontracting for production of war material. E. R. Stemper, who has been director of purchases for many years, will remain in a consulting capacity.

MAKE YOUR VENTILATING JOBS Peerless Electric

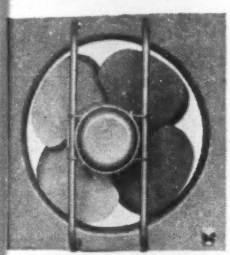
and get the benefits of 50 years' experience in building quality motors and electrical apparatus

- 26 BELT DRIVE SIZES
- 12 DIRECT DRIVE SIZES

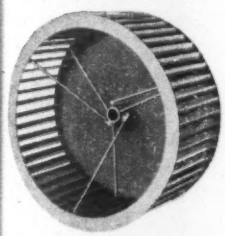




Peerless Electric
VENTILATING BLOWERS
SIZES 12" to 48"



Peerless Electric
Type PVS Deluxe Fan



Peerless Electric
BLOWER WHEEL
Forward curve blower wheel full welded construction carefully balanced for smooth, quiet rotation.

Peerless Electric VENTILATING SETS

Belt drive units are powered with ½ to 7½ h.p. motors; direct drive units ½ to 2 h.p. motors. All units supplied with or without weatherproof hood covers for drive mechanism.

NO matter what your needs—for small jobs or large jobs—it will pay you to make Peerless your headquarters for ventilating equipment. More than 50 years of experience have given Peerless Electric the "know-how" that enters into the design and manufacture of Peerless Electric fans, blowers and ventilating equipment, entirely in our own modern plant. Peerless Electric is small enough to study your needs and special applications intimately—large enough to supply your complete requirements.

Write us for details on any standard Peerless Electric ventilating equipment, or tell us your special problems.

THE Peerless Electric COMPANY
ESTABLISHED 1893 • WARREN, OHIO
MOTORS • FANS • BLOWERS

READY-POWER ENGINED REFRIGERATION

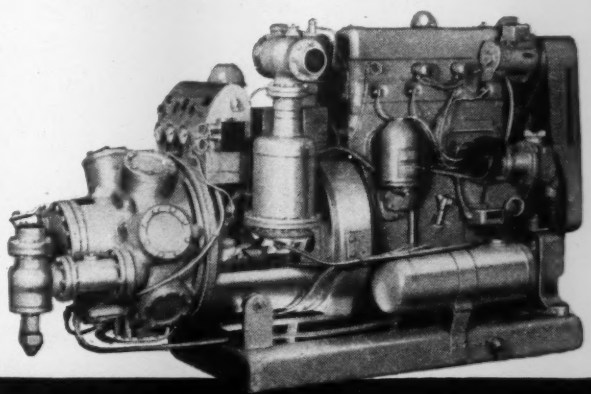
THE ANSWER TO 2 MAJOR AIR CONDITIONING PROBLEMS

1. NO THREE-PHASE POWER REQUIRED

Here is the ideal automatic air conditioning unit for any building located where power facilities are not adequate or where 3-phase commercial power is not available. The unit consists of a modern, radial compressor that is driven by a direct-connected International industrial-type engine. Ready-Power Engined Refrigeration Units are designed and built to give highest quality air conditioning.

2. LOW COST FUELS MEAN LOWEST OPERATING COST

Ready-Power Engined Refrigeration Units use cheap fuels such as natural gas, butane, gasoline or Diesel; therefore operating costs are extremely low. In many cases the costs of air conditioning with Ready-Power are but fractions of what users normally expect to pay. For installations requiring 10 to 75-ton air conditioning load. Write The Ready-Power Co. for complete information.



THE READY-POWER CO.
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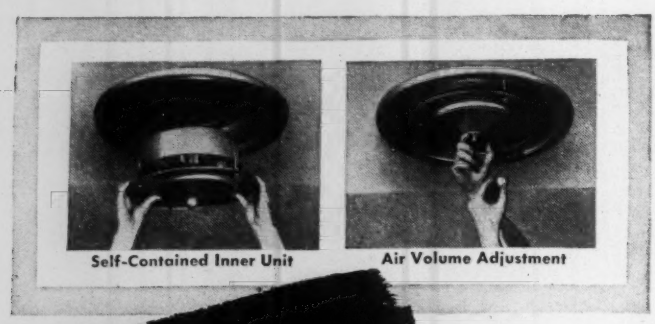
Self-Contained Inner Unit

Air Volume Adjustment

FAST INSTALLATION

"Custom-made" air pattern with Kno-Draft Adjustable Diffusers in Champion Spark Plug Plant

PROBLEM: A ceramic kiln radiates heat on one side and a cold window-filled wall creates drafts on the other. The space is long and narrow with low head room. The occupants work in short sleeves. Unusually large volumes of supply air have to be brought in to combat the heat of the kiln. Space for duct work above the room is limited so that diffusers have to be spaced much closer than usual. The supply air stream has to be directed slightly above the horizontal and evenly distributed so that no drafts are felt by the personnel.

SOLUTION: Kno-Draft Adjustable Air Diffusers were chosen because of their ability to control air direction, volume and throw. A "custom-made" air pattern was created which thoroughly mixed room and supply air, maintained uniform temperature throughout and eliminated drafts, hot spots, and cold spots. *Installation* was fast. Kno-Draft self-contained inner units cut installation time up to 50%. *Balancing* was fast. CFM readings were taken directly with Velometer. A twist of the wrist changes air volume on each diffuser.

Free — New handbook on Air Diffusion

It contains the latest engineering data on air diffusion and is profusely illustrated with charts, photographs, sketches and dimension prints that simplify the selection, application, location, assembly, erection, testing and adjusting of Kno-Draft Adjustable Air Diffusers. It is designed to help you get top efficiency from an air conditioning system by creating "custom-made" air distribution patterns.
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Industrial Environment Text Provides Valuable Data, Is Too Comprehensive

Industrial Environment and Its Control, by J. M. DallaValle deals with engineering methods for the control of health hazards in factories. It is "designed for industrial hygienists, industrial physicians and surgeons, ventilating, mechanical, and sanitary engineers, public health officers, industrial plant managers, labor union officials, and all others interested in the control of health hazards in industry."

The book is very comprehensive and it appears to me to try to cover too much in too simple and brief a fashion. The problems treated and their solution are not simple and I doubt if you can properly cover noise, vibration, and fatigue in 10 pages, nor can you cover psychrometry in seven pages.

As a quick reference book, there may be some value in having in one place chapters which treat these various subjects—atmospheric environment, thermal environment, illumination and radiant energy, and then go into the industrial ventilation problem which includes measurement of air flow, design of exhaust hoods and exhaust systems, and industrial dust collection. The book collects and reprints the material on hoods, ducts, etc. for handling various dusts, which data are useful although incomplete.

The thermal environment material is taken from some of the work of the American Society of Heating and Ventilating Engineers but is not brought up to date. For example, there is still the inclusion of the old fallacy of shock when going from a warm to a cooler room, and the old table from the 1943 Guide on the desirable temperatures running from 68 to 73 F effective temperature, depending on outdoor dry bulb. This table is in error and has not been included in the recent Guides. Any new book of this kind should be extra careful in seeing that the material taken from other sources refers to the current data of that source.

There is no question but what the problems and trends covered in the first chapter are worthwhile and emphasize the importance of the approach that occupational disease or hazard associated with bad air can be and must be corrected. Mr. DallaValle says " . . . it is significant that many occupational diseases bid fair to become medical history within a few years, not through immunization or treatment, but solely through control of their causes . . ." It certainly is *significant* that for half a century all of the emphasis has been on trying to protect man from contaminated food and water and nothing has been done about his air.

There is an appendix called General Principles of Industrial Sanitation which briefly covers all of the phases of industrial life. The footnote to this chapter says that the intent of the principles here stated is to highlight the minimum basic requirements; again this can be considered as a quick, thumb-nail reference but far from complete enough to be considered the text book or the authority that the publisher indicates on the wrapper. The problems of air, air environment, air for breathing, are in themselves tremendously complex and must be carefully studied if one is to get even a basic

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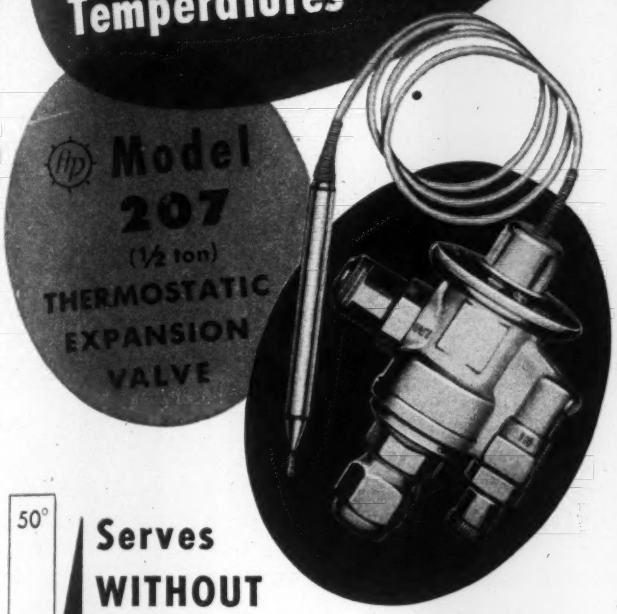

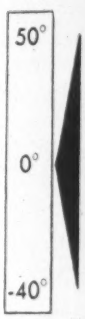

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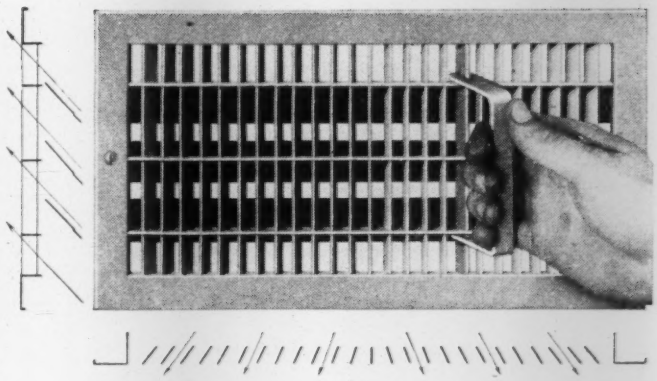
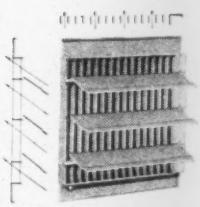


Style 321-A grille with deflecting vanes



Rear view showing adjustable deflecting vanes

ACCURATE and positive compound deflection of air flows can be readily secured with these "Fabrikated" grilles. Directional adjustments may be made when grilles are installed; or grille bars and vanes may be adjusted after installation to meet unforeseen or changed requirements.

Each grille bar and each deflecting vane is adjusted *individually* with a special two-pronged tool included with each shipment. Locking of grille bars and vanes is not required because they are held firmly in place — no vibration — no rattle.

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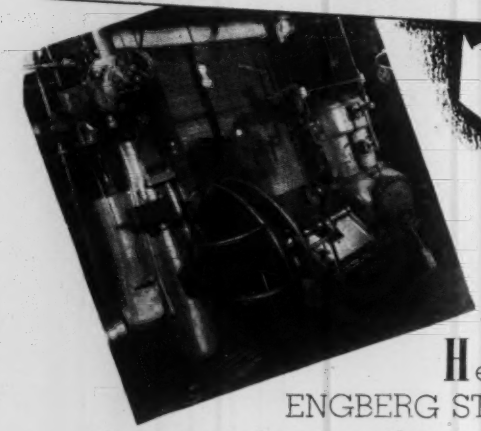
TROY-ENGBERG STEAM ENGINE

THE STANDBY DRIVE

that soon became

THE MAIN DRIVE



Here is a TROY-ENGBERG STEAM ENGINE that started out on the second team but just wouldn't stay there! This 10" stroke Type E vertical unit hooked up to a well-known ice machine was installed in SIEBEN'S BREWERY CO., CHICAGO as a standby unit for a large horizontally driven machine. After a test run it was discovered that the TROY-ENGBERG unit could carry the load with the same or less steam consumption than the larger and supposedly more economical unit. Now the TROY-ENGBERG unit carries the full load and the big unit is relegated to the standby status.

If you use steam for heating or processing, you too can probably cut steam costs, which means power costs, by installing a TROY-ENGBERG Steam Engine and using By-Product Power to drive stokers, generators, compressors, pumps or blowers.

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intelligent picture. Illumination, radiant energy, and noise are certainly big subjects. Exhaust systems and dust collection systems involve trade practice, certain code requirements, and a much better understanding of the problems of air flow.

The first sentence in Chapter Seven says . . . "the term industrial ventilation means a system of ventilation for removing air-borne contaminants . . ." While the chapter includes a casual reference to supplying or removing air by natural or mechanical means to or from any space the emphasis on *supply* is lacking although there is a rather complete calculus formula on the dilution theory which—it seems to me—might discourage and confuse the ordinary engineer who left his calculus behind when he graduated from school.

There is much of value in the book but it seems to me that it tries to cover too much and is rather a reference outline, and as such should leave out detailed formulas. If it would be a text book, it should include *more* of the up to the minute engineering and scientific data.

This 240 page (6 x 9 in.) text is priced at $4.50 and is available from the Pitman Publishing Corp., 2 W. 45th St., New York City.—Reviewed by Lester T. Avery, president, Avery Engineering Co.

Gives Schedule, Other Information for Maintenance of Electric Motors

Stating that the care of electric motors, like any other industrial procedure, should be planned, W. W. McCullough, district manufacturing and repair department, Westinghouse Electric Corp., introduces the reader to his new book, Electric Motor Maintenance, by defining the term and describing what good maintenance means. Next follows suggested schedules for inspection and repair. These are divided into operations to be performed weekly, monthly, yearly, and every two years. The importance of keeping records is stressed.

The text, intended for inspectors, mechanics, electricians, and engineers, briefly reviews some of the fundamental principles and landmarks in the development of various component parts. Subjects discussed under mechanical maintenance include the mechanical assembly of the motor, sleeve bearings and lubrication, ball, roller, and thrust bearings, current collecting devices, and airgaps. The section on electrical maintenance briefly covers cleaning, drying, and testing insulation and insulation materials. In the section on operation and application basic characteristics of the various types of motors are mentioned and additional inspection and servicing information is given.

Essentially of a practical nature, this 126 page (5½ x 8¼ in.) text is clothbound and is published by John Wiley & Sons, Inc., 440 4th Ave., New York City. The price is $2.00.

Other Books and Reports Received

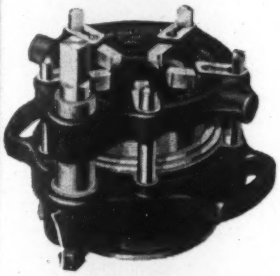
Natural Tangents to Eight Decimal Places, by R. B. Huey. Published by Parker & Co., 241 E. Fourth St., Los Angeles 13, Calif. This is a 6¾ x 10½ in. book containing 630 pages listing tangents for every second of arc from 0 to 90 deg. The price is $15.00.

At Last! ...a much easier-to-handle 2½" to 4" geared Pipe Threader

Quick-set workholder — fully enclosed gear.

New RIDGID No. 4P

● Now you can have a 2½" to 4" geared threader you can handle with unusual ease — carry it, put it on pipe; it's got handles you can get hold of and control it with. That's only one of the many RIDGID features of this new 4P die stock that you'll enjoy.



Balanced Loop-handles. Easy to pick up and put straight on pipe — won't slip or tip over. Easy to carry. . . .

Quick-setting Workholder. Mistake-proof — sets to size *before* you put it on pipe, no bushings. . . .

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Day after day and year after year, Layne Well Water Systems continue to produce the 40 tons of water required in the making of each ton of paper. It is these same kind of well water systems that Layne builds for railroads, petroleum plants, packing houses, breweries, air conditioning installations, chemical plants, irrigation projects, municipalities—or in any situation where huge quantities of water must be produced at extremely low cost.

No other well water systems are as widely used or highly praised as those bearing the name Layne. For literature, address Layne & Bowler, Inc., General Offices, Memphis 8, Tenness.

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MEETINGS & CONVENTIONS

Third Annual Home and Building Exposition, June 10 to 20, Pan-Pacific Auditorium, Los Angeles—the show is owned by the Los Angeles chamber of commerce and 14 industry associations of southern California.

National Association of Building Owners and Managers, 134 S. La Salle St., Chicago, Ill.—1948 annual convention, June 14 through 17, Hotel Jefferson, St. Louis, Mo.

I-B-R Short Course in Heating, sponsored by the University of Illinois and the Institute of Boiler and Radiator Manufacturers. To be held at the university June 15 through 17, it is a three day short course for the study of hot water and steam heating systems. Address requests for information to H. C. Rountree, assistant director, Division University Extension, room 118A, Illini Hall, University of Illinois, Champaign, Ill.

Students will be split into two groups, beginners and advanced, and will attend separate classes for individual hot water and steam heating problems. Lectures for all students will be given on heat loss calculations, blue print reading, fuels and combustion, panel heating problems, and baseboard radiation.

R. E. Ferry, general manager of I-B-R, will address the group at a dinner meeting June 16. L. N. Hunter, vice president and director, National Radiator Co., will also speak to the group.

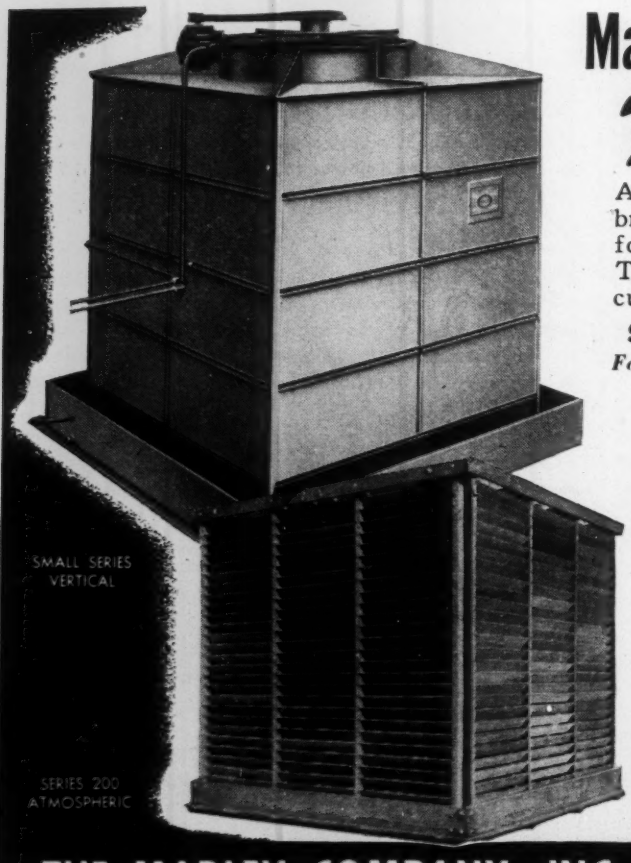
Stoker Manufacturers Association, 307 N. Michigan Ave., Chicago 1, Ill.—1948 annual meeting which was to be held at White Sulphur Springs, June 17 to 19, has been cancelled and instead, arrangements have been made to hold a one day business conference at the Edgewater Beach Hotel, Chicago, Friday, June 18th, with a technical session the following morning.

American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York City—1948 semi-annual meeting, June 21 to 23, Mount Washington hotel, Bretton Woods, N. H. See the ASHVE Journal Section in this issue for details.

American Society for Testing Materials, 1916 Race St., Philadephia—annual meeting, Book-Cadillac hotel, Detroit, June 21-25.

Over 300 technical sessions will be held, and throughout the week, the Society's Eighth Exhibit of Testing Apparatus and Related Equipment will be in progress. The exhibits will be at the Book-Cadillac Hotel, the main registration headquarters. Auxiliary headquarters will be set up at the Statler and the Detroit-Leland hotels where committee meetings and some sessions will be held.

The many technical sessions will include a symposium on tests of ball bearing greases and a panel discussion on corrosion of pressure vessels. On Friday, the last day of the meeting, discussion of fuels, fire tests, building construction, and mineral oil will be included.



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Atop a building . . . on the ground . . . obstructed from breeze . . . in open, unobstructed site . . . every location for a cooling tower constitutes an individual problem. The complete Marley line includes a unit that accurately fills any requirement.

SMALL SERIES VERTICAL INDUCED DRAFT TOWERS
For Controlled Performance or to Enhance Architectural Design
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- May be readily expanded for greater capacity.

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Marley Patented Spray Nozzles for cooling towers, processing and air washing provide maximum distribution at lowest pressure. Absolute minimum of maintenance is necessary.

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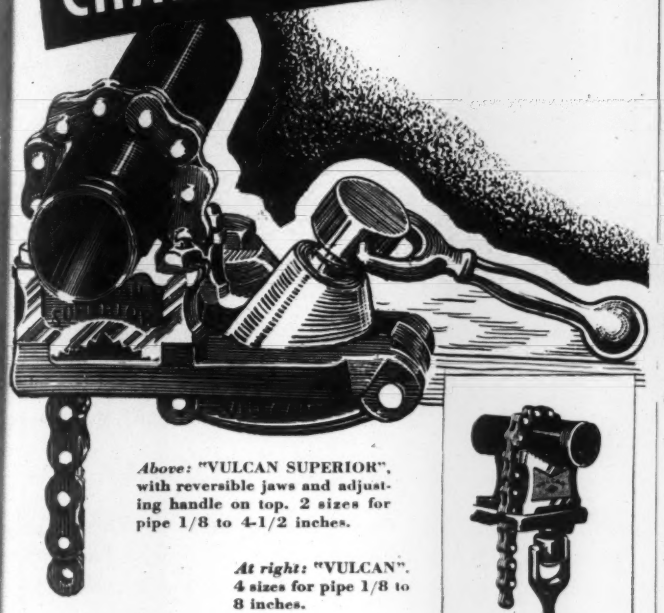
"VULCAN SUPERIOR" CHAIN PIPE VISES



Above: "VULCAN SUPERIOR", with reversible jaws and adjusting handle on top. 2 sizes for pipe 1/8 to 4-1/2 inches.

At right: "VULCAN". 4 sizes for pipe 1/8 to 8 inches.

● Williams "Vulcan Superior" is an improved vise featuring "over-head" adjustment handle ... making it easier and faster to use. Jaws are reversible providing double the service of conventional vises. It has larger capacity too. Each vise takes pipe 1/2" larger than other chain pipe vises of similar size. Like all Williams tools it is fully guaranteed. Sold by Industrial, Plumbing & Oil Field Distributors everywhere.

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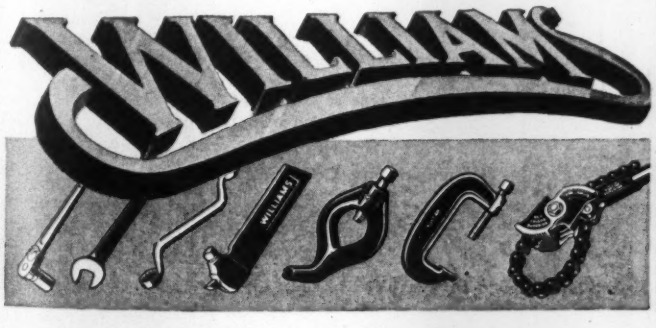

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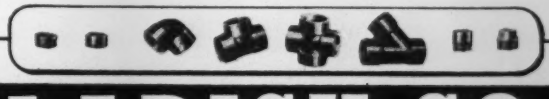

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Meetings & Conventions (Cont.)

National Warm Air Heating and Air Conditioning Association, 145 Public Sq., Cleveland—midyear convention, June 29-30, Edgewater Beach hotel, Chicago. Midyear convention of the National Heating Wholesalers Association, 2130 Keith Bldg., Cleveland 15, will be held at the Edgewater Beach on June 28.

Fred Smith, executive vice president, the Foremen's league, will address the warm air convention at a luncheon meeting on June 29th. His subject will be Management—The Responsibility for Leadership.

National Catholic Building Convention and Exposition, Inc., 185 N. Wabash Ave., Chicago, Ill.—building convention and exposition, June 30, July 1 to 3, Stevens hotel, Chicago.

The 2nd International Store Modernization Show, July 6 to 10, Grand Central Palace, New York City. Invitations for admission can be obtained from exhibitors or from the Store Modernization Show, 40 E. 49th St., New York 17, N. Y.

The Third National Instrument Conference and Exhibit—sponsored by the Instrument Society of America, 1117 Wolfendale St., Pittsburgh 22, Pa. To be held September 13 to 17, Philadelphia Convention Hall.

American Gas Association, 420 Lexington Ave., New York City—1948 convention, the week of October 4, Atlantic City.

American Chemical Society, Chicago section, 513 S. Wabash Ave., Chicago 5—Fifth National Chemical Exposition, October 12 through 16, Chicago Coliseum.

National Association of Housing Officials, 1313 E. 60th St., Chicago 37—annual meeting, October 13-16, Olympic hotel, Seattle.

The Third Annual Exhibit of Building and Maintenance Products will be held concurrently.

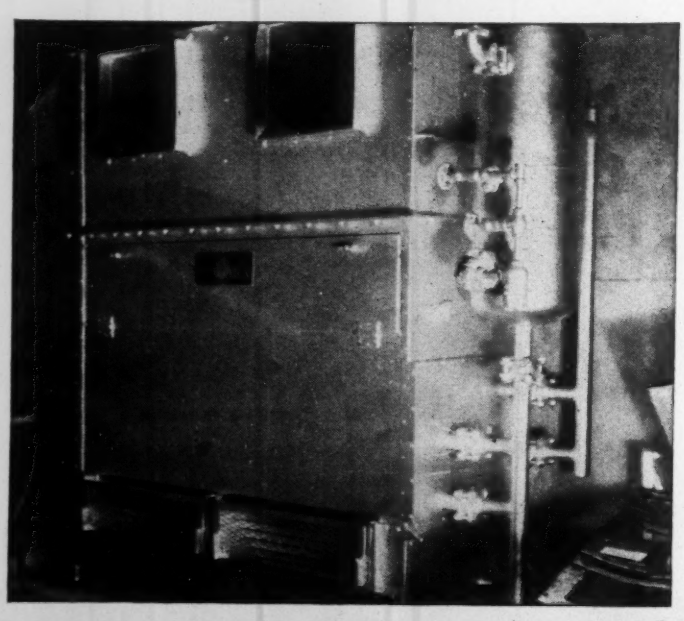
American Welding Society, 39 W. 39th St., New York City—annual meeting, October 24-29, Bellevue-Stratford hotel, Philadelphia.

American Society of Refrigerating Engineers, 40 W. 40th St., New York City—1948 annual meeting, December 6 to 8, Hotel Statler, Washington, D. C.

Ninth International Heating, Ventilating, and Air Conditioning Exposition, January 24 to 28, 1949, International Amphitheatre, Chicago—under the auspices of the American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York City.

Sixth All-Industry Refrigeration and Air Conditioning Exposition, the week of November 14, 1949, Atlantic City—sponsored by the Refrigeration Equipment Manufacturers Association, 1107 Clark Bldg., Pittsburgh 22, Pa.

NIAGARA "NO-FROST" METHOD GIVES INCREASED FROZEN STORAGE CAPACITY



Successful Niagara NO-FROST Method installations in chilling of fresh foods and quick freezing of fruits, vegetables, fish, meats, and ice cream hardening prove many advantages:

- No frost or ice at any time . . . increases capacity and lowers cost. There is no loss of capacity during frost accumulation and no interruption for defrosting.
- Versatility for freezing or chilling bulk or packaged products establishes lower initial cost.
- Narrower spread between head pressure and suction pressure assures lower power consumption by refrigeration machinery.
- Rapid chilling or freezing and accurate control of temperature assures finest quality in any food.

Write for Bulletin 83-HP

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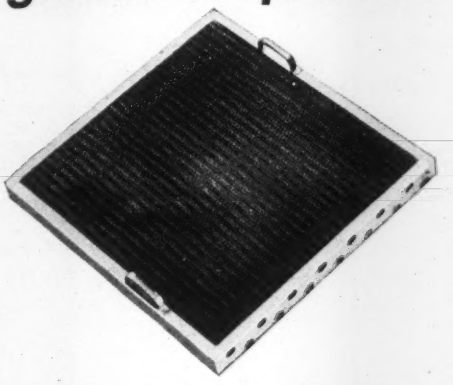
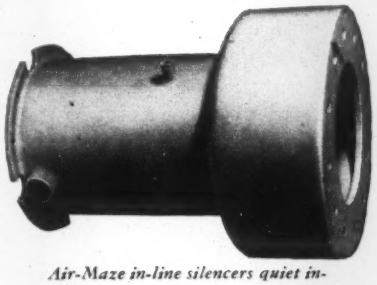
From capturing grease droplets...



Air-Maze Greastop filter panels prevent fire hazards in kitchen ventilating systems.

to hushing air for superchargers...



Air-Maze in-line silencers quiet intake noise on supercharged engines.

put your problem up to Air-Maze!

To keep fire-causing grease droplets out of commercial kitchen ventilating systems is one problem—to blanket out undesirable air intake noise on supercharged engines is another. But both jobs involve air. And there's a special Air-Maze product *engineered* to solve each of them—just as there are Air-Maze filters designed to solve scores of filtration problems involving air or liquids.

WHAT'S YOUR FILTERING PROBLEM? Whether you build or use engines, compressors, hydraulic equipment, lubricating or ventilating systems, *or any device using air or liquids*—the chances are there is an Air-Maze *engineered* filter to serve you better. Write Air-Maze Corporation, Cleveland 5, Ohio.

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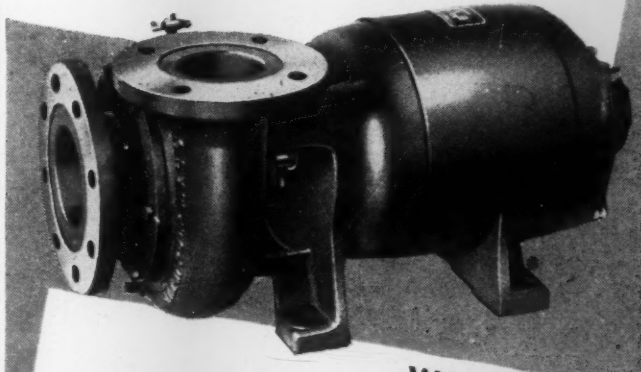
where EFFICIENCY!...ECONOMY! ...and DEPENDABILITY!...count!

A WEINMAN PUMP IS THE HEART OF THE AIR-CONDITIONING SYSTEM!

What greater proof of quality can any pump offer than its on-the-job performance record? . . . and the WEINMAN pump record for day in and day out *Dependability!* . . . *Efficiency!* . . . and *Economy!* . . . as the heart of countless air-conditioning systems, is a record to convince you that *Weinman Pumps Are "Right" For The Job* . . . made right by centrifugal specialists who know air-conditioning pump requirements.



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Especially adapted for circulation of chilled water or brine. Sizes to provide capacities of 10 G.P.M. to 1000 G.P.M., heads to 500 ft., and motor ratings ranging from ½ hp. to 30 hp.

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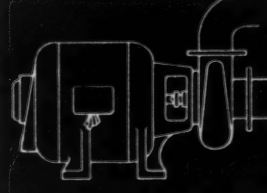

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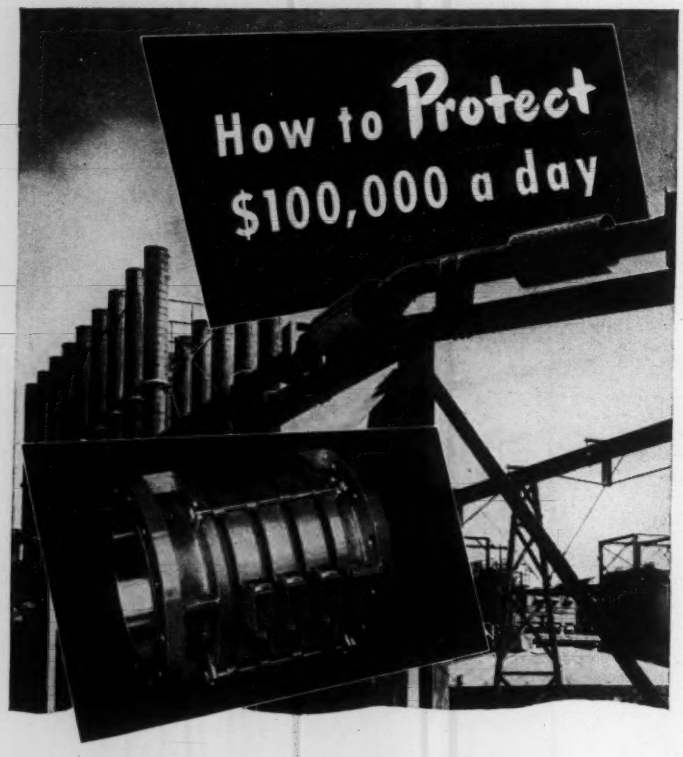


THE ZALLEA Stainless Steel Expansion Joint shown above is one of many installed in the outdoor steam distribution line of a large eastern industrial plant which has a production capacity of over $100,000 per day.

The failure of one of these joints would halt production and result in costly delays. But these Zallea Expansion Joints, installed years ago, are still doing a "bang-up" job today of "protecting" daily production by absorbing movement due to contraction and expansion of the steam line.

Long life, dependability and trouble-free performance are built into Zallea Expansion Joints to make them last, even in the severest service. There's no packing to deteriorate . . . no maintenance required. Be *sure* of your expansion joints . . . specify Zallea.

Zallea Corrugated Expansion Joints are made in non-equalizing, self-equalizing and duo-equalizing types, depending upon operating conditions . . . of copper, stainless steels and corrosion-resistant alloys . . . up to 72" diameter for a very wide range of temperatures and pressures. Send NOW for complete information contained in the new Zallea Catalog 47. ZALLEA BROTHERS, 816 Locust St., Wilmington 99, Del.

Zallea
EXPANSION JOINTS

WORLD'S LARGEST MANUFACTURERS OF EXPANSION JOINTS EXCLUSIVELY

RECENT TRADE LITERATURE

For your convenience in obtaining copies of these bulletins, see coupon on page 189. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending requests to Heating, Piping & Air Conditioning.

Air Velocity Meter

No. 6855—The new "Alnor Velometer Jr.," recently announced by Illinois Testing Laboratories, Inc., 420 N. LaSalle St., Chicago, is said to be a portable, low priced, direct reading instrument giving instantaneous and precise measurements of air velocities in unrestricted areas. It is described and illustrated in a new leaflet.

Automatic Control and Regulating Valves

No. 6856—Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill., has released a condensed price list bulletin (No. 48-2) on its automatic control and regulating valves. Every valve is illustrated and a brief description is given with each listing. Technical data for selecting the proper water regulating valve for water cooled condensing units are included.

Blower Wheels

No. 6857—Bulletin 100, released by Revcor, 3227 W. Carroll Ave., Chicago 24, Ill., describes blower wheels of steel, brass, aluminum, and stainless steel. Dimensions (including those recommended for sheet metal housings), performance data, and information on special hubs are included.

Brass Fittings

No. 6858—Handy Data on Brass Fittings is the title of a folder recently published by Imperial Brass Mfg. Co., 1200 W. Harrison St., Chicago 7, Ill. It describes the various types of fittings offered by the company including two-way and three-way shutoff valves and drain cocks. In addition to price listings, cross section views of the various joints and disassembled views are shown.

Coal Pulverizing and Burning Equipment

No. 6859—Coal pulverizing and burning equipment for industrial furnaces and for steam generation is discussed in a bulletin issued by Babcock & Wilcox Co., 85 Liberty St., New York 6, N. Y. The development and application of this equipment is discussed, and the text is illustrated with diagrams and photographs.

Condensate Return Units

No. 6860—A new 10 page bulletin (No. 201) describing type U packaged condensate return units has been issued by Roy E. Roth Co., Rock Island, Ill. This booklet provides general catalog data, descriptions, specifications and prices on the company's units, parts, and accessories.

Continuous Indicating and Recording Flue Gas Analyzer

No. 6861—Davis Emergency Equipment Co., Instrument Div., 45 Halleck St., Newark 4, N. J., has released a new four page bulletin (1136A) on its continuous indicating and recording flue gas analyzer. In addition to explaining how this unit analyzes the composite of stack gases and indicates deviation from ideal combustion conditions, the bulletin covers the effect stack gases have on combustion efficiency of fuel. Also included is information on how to eliminate incomplete combustion and excess or insufficient air. How continuous analysis of stack gases insure proper combustion is also covered.

Drawing, Forming, and Welding Aluminum

No. 6862—A brief article on selecting aluminum for drawing and forming operations is the lead off item in the Vol. 1, No. 7 issue of the Technical Advisor, house organ of Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky. Also included is Section VII in a series of articles on welding aluminum, taken from the company's 88 page manual on this subject. Seam welding, flash welding, and brazing methods are discussed.

Forced Circulation Hot Water Heating Specialties

No. 6863—Catalog F-99, issued by Taco Heaters, Inc., 137 South St., Providence 3, R. I., covers the company's complete line of specialties for hot water heating systems. New rating tables on converters are included, as are several new products such as the "Paneltrol" water temperature regulating valve for panel heating systems. Also covered are circulators, flow valves, relief and reducing valves, storage and tankless heaters, tempering valves, and the fittings used in connection with the company's "Taco-One" venturi system.

Galvanic Corrosion

No. 6864—A new booklet discussing various aspects of galvanic corrosion has been issued by International Nickel Co., 67 Wall St., New York 5, N. Y. Prepared under the supervision of the company's corrosion engineering section, it is designed for the production man as well as the research engineer and covers some of the factors influencing galvanic corrosion.

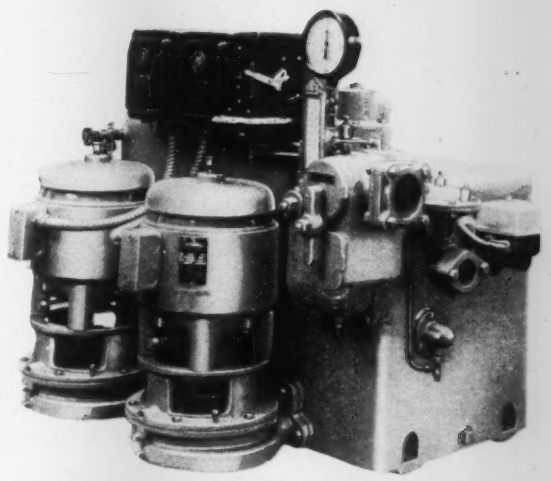
Geared Motors

No. 6865—The use of geared motor power in driving machinery and the latest models of "Syncrogear" motors are described and illustrated in a new 16 page bulletin issued by U. S. Electrical Motors, Inc., 200 E. Slauson Ave., Los Angeles 54, Calif. Advantages of using hardened gears and pinions are described, and the company's geared motors in single, double, and triple reductions for use in combination with its "Varidrive" units are also covered. Sizes range from ¼ to 30 hp and speeds range from 10 to 10,000 rpm.

Hose Fittings

No. 6866—A new catalog section on the fittings available for the diversified lines of hose offered by B. F.

Get MORE EFFICIENCY



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VACUUM HEATING
PUMPS

Special foundations are eliminated—installation is made easy—because companion flanges are furnished for return and discharge connections. 44 years of engineering "know-how" have established DOMESTIC AMES vacuum heating pumps as leaders in supplying low maintenance cost and longer service at maximum rated capacity. Single or duplex units with Standard or Special capacity. Write Dep't. H for FREE catalog.

"If it's a job for a pump - it's a job for Domestic"

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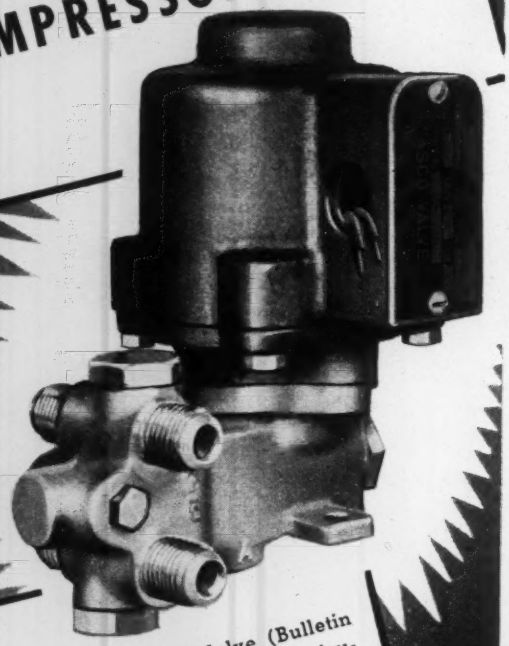
ASCO SOLENOID VALVES

IDEAL "Unloading" CONTROL FOR FREON COMPRESSORS...



It's the ASCO 3-Way Solenoid Valve (Bulletin 8304) designed, as our catalog reads, "particularly for compressor unloading service handling refrigerants such as freon and methyl chloride: size 1/8 in. pipe with 1/2 in. SAE male flare; pressure up to 250 psi."

The words "designed particularly for" mean just that but back of this special work is an experience in the design and manufacture of solenoid valves that goes back some forty years. It means that we were able to bring to that problem, the cumulative knowledge gained by working on thousands of solenoid valve applications.

Our engineering department is at your service either (1) to help select the proper valve for your requirements from the present extensive line of ASCO Valves or (2) to design a special valve should you need one. In writing, please give all operating requirements and conditions.

When in need of Automatic Transfer Switches, Remote Control Switches, Contactors, Relays, and Specialized Electromagnetic Controls, come to us.

Automatic Switch Co.

389 Lakeside Avenue Orange, New Jersey

Goodrich Co., Akron, Ohio, is now off the press. Pointing out that proper attachments increase the service life of the product and insure maximum safety, the section provides recommendations for attaching couplings to hose. Also included is a table showing recommended couplings for different types of hose and a detailed discussion of coupling threads and their functions.

Intake Filters

No. 6867—An eight page bulletin, published by Dollinger Corp., 11 Centre Park, Rochester 3, N. Y., covers the line of "Staynew" intake filters for internal combustion engines, compressors, blowers, motors, and generators. Sizes, dimensions, and capacities of these dry type filters are tabulated.

Ion Exchange Material

No. 6868—National Aluminate Corp., 6220 W. 66th Pl., Chicago 38, Ill., has announced the availability of a new booklet on ion exchange principles, properties, and uses which includes technical data on various materials. In addition to a general discussion of ion principles, the booklet provides data on each of the company's commercially available types of ion exchange materials. According to the company, the information is designed to equip technical men to predict performance under specific operating conditions. Brief notes on a number of chemical manufacturing and processing operations employing these exchangers are also included.

Mechanical Draft Cooling Towers

No. 6869—Binks Mfg. Co., Water Cooling Div., 3126-36 Carroll Ave., Chicago 12, Ill., has announced two technical bulletins on its mechanical draft cooling towers. Bulletin 36 describes recent design changes on its type 2K-S (spray filled) induced draft cooling tower which is said to have many commercial water cooling applications and which is available in capacities from 150 to 645 gpm. Bulletin 37 covers the type 2K-W (wood filled) induced draft cooling tower which is designed for many types of water cooling operations in refrigeration, air conditioning, manufacturing, and other allied applications where interior decking is needed for the distribution and break-up of the water. This unit is supplied in 20 standard sizes with capacities ranging from 150 to 495 gpm.

Mechanical Roof Ventilators

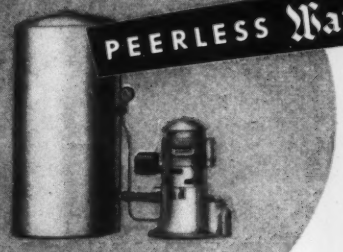
No. 6870— "Iron Lung" roof ventilators, utilizing propeller fans, are designed to exhaust large volumes of air from industrial buildings. Incorporating dampers which are closed when the fan is not in operation, the ventilators are available in various sizes for handling air quantities from 3720 to 40,000 cfm. They are covered in a new bulletin issued by Powermatic Ventilator Co., 4019 Prospect Ave., Cleveland 3, Ohio.

Mechanized Process of Hot Dip Galvanizing

No. 6871—Perfection of a completely mechanized process of hot dip galvanizing has been announced by Rheem

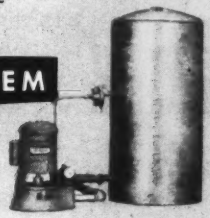
FOR SHALLOW WELLS

PEERLESS Water King



FOR DEEP OR SHALLOW WELLS

PEERLESS JET SYSTEM



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FOOD MACHINERY CORPORATION

Factories: Los Angeles 31, California; Quincy, Illinois; Indianapolis, Ind.; District Offices: Chicago 40, 4554 N. Broadway; New York 5, 37 Wall Street; Atlanta Office: Rutland Bldg., Decatur, Georgia; Dallas 1, Texas; Fresno, Calif.; Los Angeles 31, Calif.

PEERLESS PUMPS

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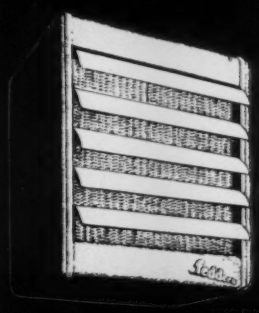
How FEDDERS UNIT HEATERS are Built to Give SERVICE

Look in Classified Telephone Directory or write for Catalog



● LOOK at the Fedders unit heater cabinet and you will find:

—Die-formed sections for strength and rigidity

—Electrically welded mono-piece simplicity

—Handsome design and morocco-like finish suitable for both commercial and industrial installations

Fedders cabinets provide a rugged support protecting heating element against piping strains.

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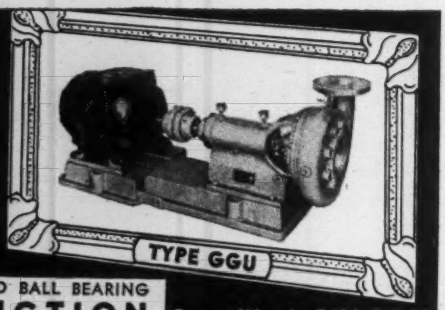
PORTRAIT of a FINE PUMP "by Aurora"



DOUBLE OUTBOARD BALL BEARING
SIDE SUCTION CENTRIFUGAL

Capacities to 1400 G.P.M.
Heads to 190 Ft.

Wherever liquids play an important or incidental part in industrial processing, this Aurora Type GGU serves perfectly and with lasting efficiency. Because the casing is separate from the bearing bracket, the need for special metals to handle corrosives is minimized. Discharge may be placed in any of eight positions. We commend these pumps for high satisfaction and economy.

APCO TURBINE-TYPE PUMPS

For high pressure, small capacity requirements (up to 150 g.p.m.). Wide operating range characteristics. No metal to metal contact. Handles non-lubricating liquids without wear. Only one moving part — the Impeller.

AURORA CENTRIFUGALS
for Every Pumping Job
HORIZONTAL SPLIT CASE
Single & Two Stage
SIDE SUCTION
VERTICAL
NON CLOG
SUMP
MIXED FLOW
DEEP WELL
TURBINES
SPECIAL DESIGN

Write for CONDENSED CATALOG "M"

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AURORA Always PUMPS
AURORA PUMP COMPANY
80 Loucks Street, AURORA, ILLINOIS

Mfg. Co., 570 Lexington Ave., New York 22, N. Y. T process is used on steel shipping containers as well tanks manufactured by the company. According to manufacturer, this type of galvanizing provides lo service life. The new process is described and illustrat in a new four page bulletin.

Packaged Boiler Feed Units

No. 6872—Roy E. Roth Co., Rock Island, Ill., has pu lished a new six page bulletin (No. 202) describing packaged boiler feed units. General catalog data, d scriptions, specifications, and prices are included.

Preventing and Removing Scale, Rust, and Corrosion

No. 6873—The "Beco Cell," developed by Brow Electro Co., 17 N. Dewey St., Oklahoma City 2, Okla., i self contained unit designed for installation in any h system. According to the company, it prevents and re moves rust, scale, and corrosion by the generation of slight negative current which attacks and neutralize foreign particles, rendering them incapable of ion attraction. It is described in a series of new leafle which also illustrate various applications in boilers, h water generators, radiators, etc.

Radiant Insulation

No. 6874—"Infra" insulation is an aluminum insula tion which is designed to absorb and emit only 3 perce radiant heat. It is made-up to form four series of i

IMPORTANT

TO DISTRIBUTORS AND DEALERS

Production now permits expanding the distribution of GR Air Conditioning and Lipman Automatic Refrigeration.

This means a truly great opportunity for qualified distributors and dealers in handling these two well-known lines.

Consider some important facts about the equipment and the company behind it:

ESTABLISHED REPUTATION — Lipman Commercial Refrigeration has been well known for 30 years, GR Air Conditioning for 25 years.

WELL ENGINEERED — All the "bugs" are out. Units are slow speed, heavy duty, rugged — perform well, stand up in service.

LINES COMPLETE — Lipman machines are available in ¼ thru 40-hp. capacities — air and water cooled — ammonia, freon-12 or methyl chloride refrigerants. The GR line includes range of self-contained and remote air conditioning units for commercial requirements.

PROMPT DELIVERY — Production facilities now permit prompt delivery — immediate shipment of some units.

SALES HELPS — Stepped-up national advertising . . . useful sales literature . . . local promotion program you can tailor to fit your needs . . . engineering service to help you . . . real cooperation at every point.

For complete information on handling the profitable Lipman and GR lines, write, wire or call immediately. If you like, just write your name and address on the border of this ad, tear it out and mail it to Dept. HP5 . . .

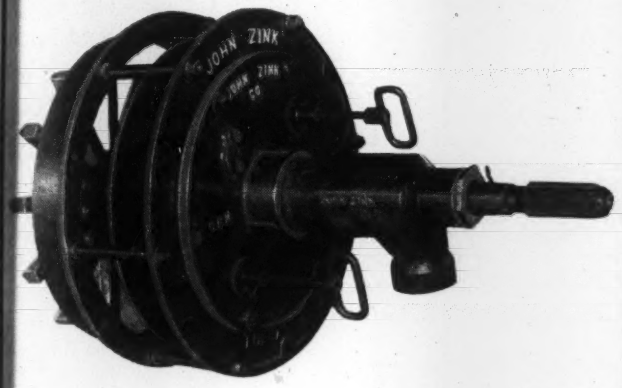
GENERAL REFRIGERATION

DIVISION
YATES-AMERICAN MACHINE COMPANY
BELOIT, WISCONSIN

GR AIR CONDITIONING

Lipman AUTOMATIC REFRIGERATION

JOHN ZINK



SERIES CBM-S
LOW PRESSURE
BURNER

A Combination Gas and Oil Burner Featuring the Steamizer Type MTB Oil Gun

It is particularly adapted to Industrial Heating and Power Boiler Installations

•

The John Zink BI-MIX GAS BURNER has been combined with a Steamizer type Oil Burner to produce a highly efficient COMBINATION oil and gas burner. It provides approximately the same heat pattern when firing either or both fuels in combination. Uses steam or gas for atomization of fuel oil.

By adding a mixer to the CBM-S Burner it will operate on gas pressures in excess of 5 lbs.

— Write for Literature —

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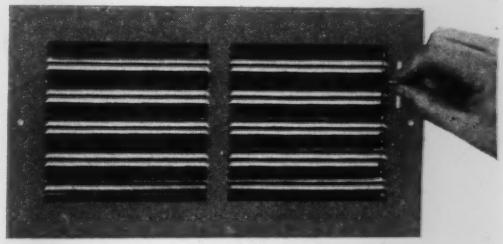
You Can Get It QUICKER from STEWART

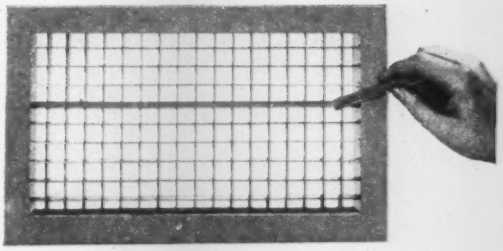
HIGH VELOCITY OUTLETS
Engineered for Dependability--Flexibility

The Stewart line of high velocity outlets, scoopaires, wall and baseboard registers meet the most exacting requirements of air conditioning, heating and ventilating systems. Special types and sizes speedily produced to order —all standard equipment promptly delivered.

Style 94
Wall or Ceiling Register
Multiple Damper Lever Regulated



Size 94 is designed for wall or ceiling use, with horizontal face bars which are down deflecting. The rear bank of the Style 94 Register serves to regulate the air supply and to deflect air into the outlet. Lever operated multivalves control the degree of opening.



Style DDH Double Directional 2 Banks —adjustable face and rear bars.

Style DDH Stewart High Velocity outlet with "All-Way-Selective-Air-Throw" produced by means of a removable key which can adjust any bar at various angles without affecting the setting of any other bar. Horizontal face bars and vertical bars in rear, are both adjustable for any combination of air direction desired.

Send for literature and prompt quotation on your requirements.

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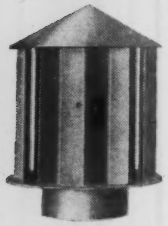
Breidert Air-X-Hausters

(PAT. NO. 2269428)

Breidert Air-X-Hauster Chimney Top makes fireplace draw better.

the first scientific improvement in roof ventilators in over 50 years!



Type B Breidert Air-X-Hauster for factories, commercial and residential buildings.

No matter which way the wind blows, no matter how bad the weather—the Breidert *provides positive exhaust.* Designed in accordance with the science of aerodynamics, this revolutionary ventilator utilizes the power of air currents coming from *all angles* to provide safe, positive ventilation without back-draft. Stationary—no moving parts to regulate or get out of order . . . no power consumption . . . compact and pleasing in appearance.

Breidert Air-X-Hauster—the first ventilator to be tested under all variable vertical and horizontal wind conditions . . . with certified ratings published.

Unlike most conventional ventilators which work efficiently only when wind strikes in a horizontal plane, the Breidert has been thoroughly tested—by U. S. Navy and Smith Emery Commercial Testing Laboratories—with the wind blowing in *all* directions. Its remarkably high performance ratings under all conditions are *certified and published.*

Many Thousands in Use all over the Country

Breidert Air-X-Hausters have found wide acceptance among architects, engineers, industrialists, farmers, home owners, ship and boat owners, U. S. Army, U. S. Navy, Maritime Commission, schools, institutions and public buildings. In every case when properly installed, the Breidert has never failed to fully meet every claim made for it!

Free Engineering Data Book, with complete specifications, sent on request. Address

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Representatives in 50 principal cities throughout the United States

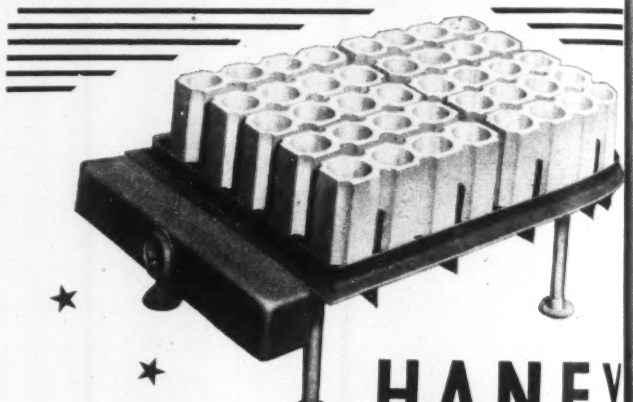
compartments as well as four aluminum surfaces. It is described and illustrated in a 16 page booklet released by Infra Insulation, Inc., 10 Murray St., New York City. Also included are data showing the relative insulating values of different materials under different circumstances. One of the features of the manual is said to be a master chart showing the *k, C, R* and *U* values of insulations in various thicknesses, weights, and densities.

Reactivation of Silica Gel by Electric Heaters

No. 6875—Control of humidity in vessels of the Navy "mothball" fleet is accomplished by the use of dynamic silica gel dehumidifying machines. How the silica gel is reactivated by the use of "Calrod" electric heating elements is briefly told in the Vol. 22, No. 1 issue of Inco, the house organ of International Nickel Co., Inc., 67 Wall St., New York 5, N. Y. The resistance wires of the heating elements are of nickel-chrome alloy and a monel sheathing is used to prevent corrosion and deterioration.

Refrigeration, Air Conditioning Equipment

No. 6876—The products, plant facilities, and manufacturing methods of the Airtemp Div. of Chrysler Corp., 1119 Leo St., Dayton 1, Ohio, are described and illustrated in a new 24 page booklet. Some of the equipment covered includes the company's sealed radial compressor, the packaged air conditioning unit, and the equipment for capacity reduction on large radial compressors.



HANEY GAS BURNERS

Haney Venturi Gas Burners are the answer to all your heating installations whether commercial, domestic, or industrial. The vertical type illustrated will afford economical, even, dependable heat with remarkable freedom from maintenance troubles. It's shipped completely assembled, ready for installation and is available for mixed, natural or LP gases.

We'll be glad to send more information on request . . . please write!

HANEY GAS BURNER AND ENGINEERS

909 South Myrtle Avenue Monrovia, California

3 DEPENDABLE SERVANTS of AIR CONDITIONING and HEATING

ALL MADE TO FUNCTION IN HARMONY BY THE SAME MANUFACTURER

FIG. R 5506

This Single-Shaft PUMP and MOTOR UNIT is an all ball bearing unit with none of the bearings in the casing. Bearings are sealed against dirt and require only one lubrication a year. Sizes with capacities from 10 to 750 G.P.M. Made in 2 types. Type R.U. with packed Stuffing Box for Cooling Tower service. Type RY with leak proof Mechanical Seal for Hot or Chilled Water Circulating.

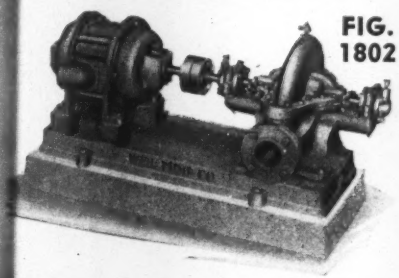


FIG. 1802

High efficiency, Double Suction Pump with Horizontally Split Case. Performs the same services as Fig. R 5506 Pump and is used where greater volume is required. Capacities from 50 to 1200 G.P.M. and heads to 250 ft. Also designed for Booster Service.

Ideal for automatically handling returns from heating systems, our "TVC" Vertical Condensation Pumps and Receivers, requiring a minimum of space, are built in capacities of 500 to 6000 sq. ft. E.D.R. against 10 to 40 P.S.I. These compact units come completely assembled, ready for final connections.

FIG. 302

Bulletins Are Available From Your Jobber or Our Factory

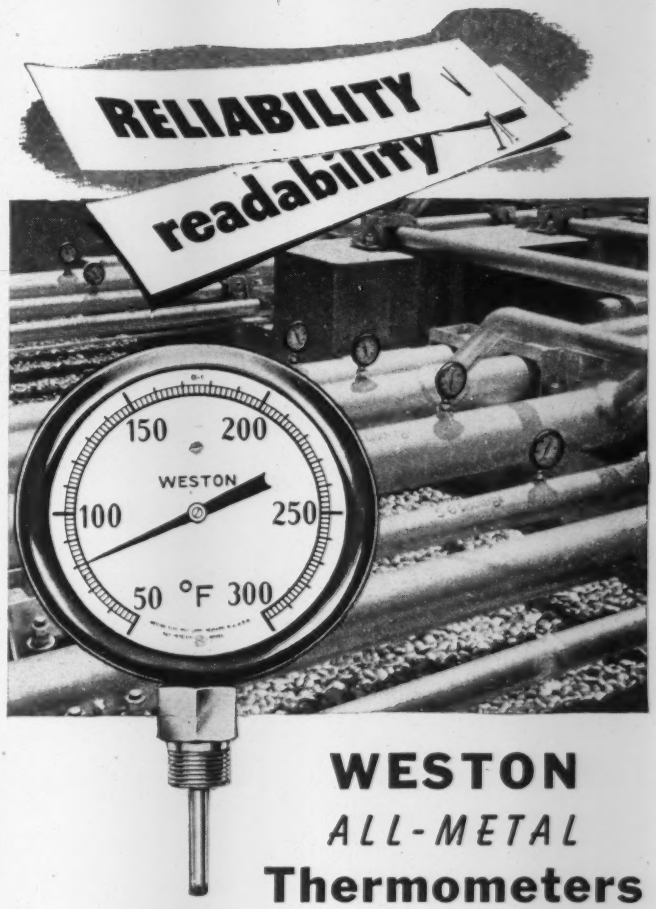
Fig. R5506, See Bulletin R5500
Fig. 1802, See Bulletin A1800
Fig. 302, See Bulletin TVC300
Other Weil Condensate Pumps—see Bulletins RC5000, TR100, SRC450, NRG200

WEIL PUMP COMPANY

Manufacturers of Centrifugal Pumps

1523 N. FREMONT CHICAGO 22, ILLINOIS

Combines the two!



WESTON *ALL-METAL* Thermometers

The legible, wide open scale on the WESTON thermometer permits you to take full advantage of its inherent, long-time accuracy. Even from a distance, readings can be made "right on the nose."

WESTON thermometers are available in a variety of types, stem lengths and scale ranges for most industrial applications. If your jobber cannot supply you, see your local WESTON representative, or write for Thermometer Bulletin . . . Weston Electrical Instrument Corporation, 648 Frelinghuysen Avenue, Newark 5, New Jersey.

★ *MAX-MIN models also available to indicate highest or lowest temperature reached.*

Weston Instruments

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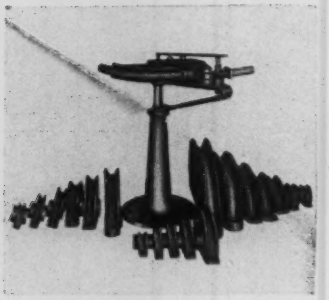
Use "American"

PIPE BENDING MACHINES . . .

for fast, dependable bends

Radiant HEAT Pipe Bending is simple, accurate with good equipment . . . hand or motor-powered machines available . . . up to 8" capacities.



The Hand-powered "American" A-30

Bends ½" to 2" standard pipe sizes to a radius of 5 to 1, up to 180°. Special radii within these sizes supplied on request. Usual radiant heat bends for 1¼" at 6" & 9" radii can be supplied. Special radius supplied on request.

- Average bend rakes only 60 seconds!
- Versatile—bends cold pipe, conduit and tube, copper and brass I.P.S. without crimping.
- 12,000 in use all over the world.

BENT ONCE — BENT RIGHT
ON "AMERICAN" PIPE BENDING MACHINES

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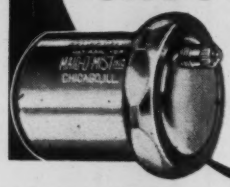
Factory and Main Office: 17 Furnace St., Poultney, Vermont

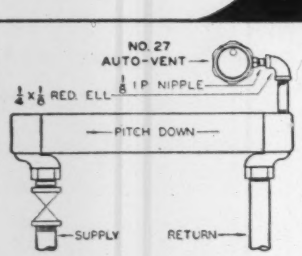
HOW TO GET **PERFECT CIRCULATION** *in Hot Water Radiators and Convectors*



AUTO-VENT AIR ELIMINATORS REALLY DO A JOB!

As air accumulates in radiators or pipes, it is AUTOMATICALLY VENTED when you use these inexpensive No. 27s. Your customer gets better heating and saves fuel.

AUTO-VENTS work equally well on convectors, baseboard radiation, panels, radiant heating coils, etc. — NO AIR CHAMBER REQUIRED. Good for pressures to 50 lbs. *See your jobber, or write us direct.*



SIMPLE AS A-B-C

Detailed drawing of No. 27 Auto-Vent Air Eliminator operating on convector radiator. Auto-Vents used on other types of jobs are just as easily installed — and no attention required.

MAID-O-MIST INC.
3215 N. PULASKI RD.
CHICAGO 41, ILLINOIS

AUTOMATIC HUMIDIFIERS FOR ALL TYPES OF HEATING SYSTEMS

Registers and Grilles

No. 6877—List prices, dimensions, types of finishes, and engineering data on double diffusion grilles, return air grilles, volume controllers, multi-shutter registers, and the new "Airfoil" louver are contained in the 1948 catalog (No. 23) issued by Titus Mfg. Corp., Waterloo, Iowa.

Rotary Pumps, Valves and Strainers

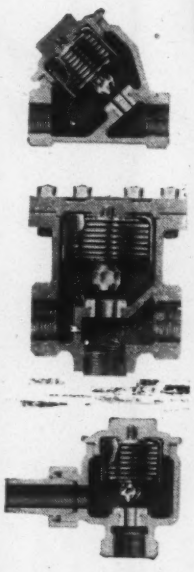
No. 6878—Caton Co., 972 Broad St., Newark 2, N. J., has issued a series of data sheets on valves, rotary pumps, and strainers. Data on sizes, capacities, and design and construction features are given on a solenoid operated flow reversing valve of stainless steel, rationing valves in 12 sizes with three-way action and with a by-pass for delivering liquids from zero to full volume, and pressure relief valves of stainless steel. Also covered are stainless steel strainers with monel baskets and 11 sizes of stainless steel rotary gear pumps having capacities from ¼ to 30 gpm.

Small Relays for Electronic Control Equipment

No. 6879—Allied Control Co., 2 East End Ave., New York 21, N. Y., has published a four page folder on its line of small relays for electronic control equipment. A detailed table enumerates the specifications of each relay, including maximum contact arrangement, contact rating current, coil operation, coil data in volt-amperes, etc.

More Service Tests Confirm that Nicholson Steam Traps Furnish...

2 to 6 Times MORE DRAINAGE Capacity

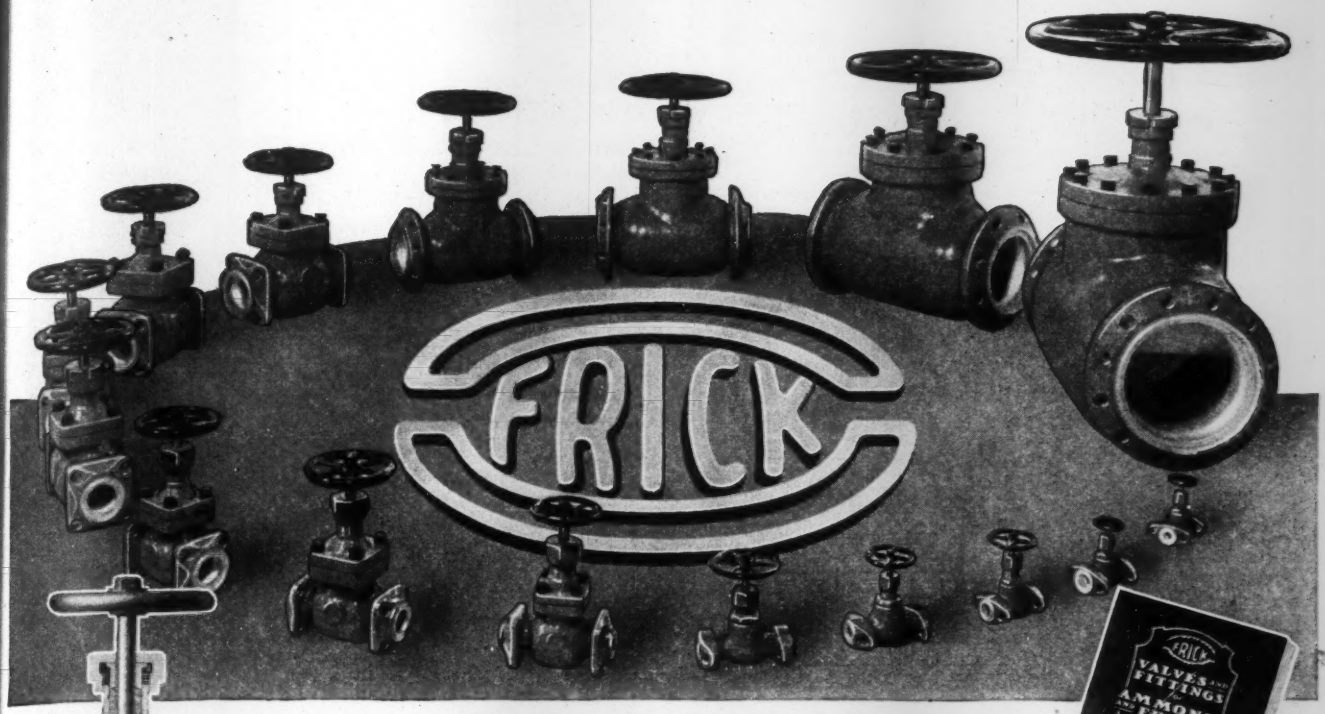


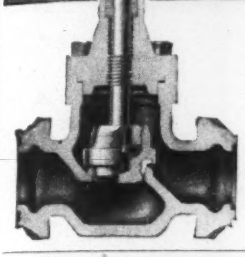
Recent comparative tests by large users of steam traps again showed the unequalled drainage capacity of Nicholson traps. The record low for steam waste was also shown. Some Nicholson features: larger valve orifice; operate on lowest temperature differential; no freeze-ups; no adjustment of valve or seat.

Put a Nicholson Trap on Test

See why leading plants are increasingly standardizing on Nicholson steam traps. We'll gladly send one for testing. 5 types for every application: size ¼" to 2"; pressure to 225 lbs. BULLETIN 1047.

W. H. NICHOLSON & CO. 198 OREGON STREET WILKES-BARRE, PA.
Valves • Traps • Steam Specialties





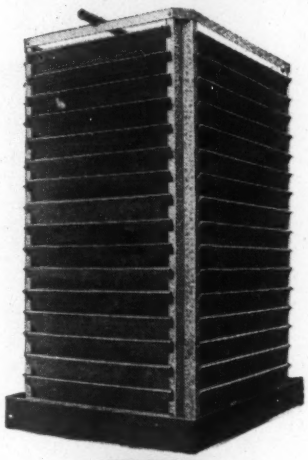
FRICK HIGH-PRESSURE VALVES

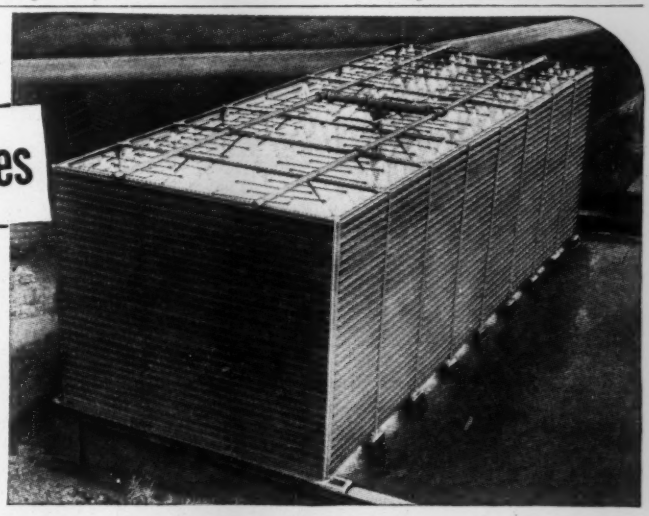
Handle loads from 300 to 1500 lb. gauge, according to size, whether using ammonia, Freon, methyl chloride or other high pressure gases and liquids, at either normal or low temperature. Only in Frick valves do you get the combination of high-angle seat, alloy-faced button, and easy re-packing, among other features which have given them the preference for generations . . . Full range of sizes, ¼" to 14". Screwed valves in sizes up to 2". Frick valves and fittings are stocked by Frick Distributors in principal cities everywhere. Ask for Catalog L; your copy is waiting. Write

FRICK COMPANY
Waynesboro, Penna.

Binks NATURAL DRAFT COOLING TOWERS

30 standard sizes





Binks Type "S" Cooling Towers are manufactured in capacity ranges from 10 to 1200 GPM. This wide range of sizes enables you to choose exactly the tower you need to fit your requirements without the needless expense of a custom-built unit. Binks Type "S" Towers are the accepted standard for cooling condenser circulating water from refrigeration and air conditioning compressors, and for cooling water-jacketed engines. They will provide low cost cooling of highest efficiency for any manufacturing process where heat is dissipated by circulating water or other fluids.

Send today

for bulletins describing the Binks Type "S" natural draft cooling towers in which you are interested.

BULLETIN NUMBERS:

30. Single Section Towers
31. Multiple Section Towers (one bay wide)
32. Large Capacity Towers
40. All Redwood Towers

Please state how tower will be used, also capacity required.

THERE'S A BINKS TOWER FOR EVERY COOLING JOB

Binks MANUFACTURING COMPANY

REPRESENTATIVES IN ALL PRINCIPAL CITIES ■ 3118-38 CARROLL AVENUE, CHICAGO, ILL.

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WHEN A FELLOW NEEDS A

FLANGE!

Here's a fellow solving his flange problems the easy way . . . by turning to the handy Phoenix Catalog where all sizes and types of Phoenix Flanges are listed in clear, easy-to-read tables. He's typical of the men who are careful buyers of flanges . . . men who keep a record of flange performance . . . men who demand safety, service and satisfaction. That's why they specify Phoenix Flanges.

PHOENIX MANUFACTURING COMPANY

Joliet, Illinois, U. S. A. △ Catasauqua, Pa., U. S. A.

BETTER

Flexible

METAL HOSE

FOR EVERY PURPOSE

• SEAMLESS—Constructed from one continuous piece of uniform thickness • PARALLEL CORRUGATION assures maximum strength and flexibility • SPECIAL ALLOYS provide resistance to heat, acid, etc., depending on requirements • AVAILABLE in wide variety of types to convey all kinds of liquids and gases • AVAILABLE in complete form as vibration reducers for Air Conditioning and Refrigeration Units.

White Eclipse-Pioneer for information.

ECLIPSE-PIONEER Division of Bendix AVIATION CORPORATION
TETERBORO, NEW JERSEY

Speed Changer

No. 6880—A new four page booklet on the "Speed Selector," designed to permit stepless, infinite ratio changes in rapid succession by a new development in planetary motion, has been released by B. F. Goodrich Co., Akron, Ohio. Utilization of two v belts and four variable pitch sheaves is said to provide any speed change from 0 to 800 rpm at constant torque. The bulletin describes the principles of operation and illustrates a number of applications.

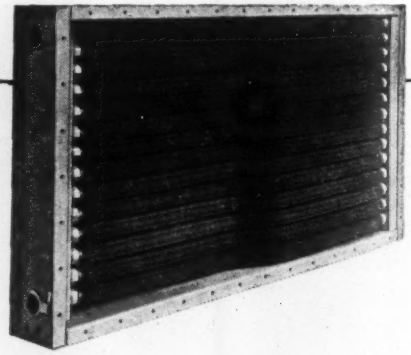
Steam Jet Ejectors

No. 6881—C. H. Wheeler Mfg. Co., Lehigh & Sedgley Aves., Philadelphia 32, Pa., describes, in Catalog No. 1462, its line of steam jet ejectors for vacuum service and for vacuum refrigeration. A wide variety of ejectors, designated as "Tubejets," is described and illustrated, including one, two, three, four, and five stage units. Also covered are the company's direct contact condensers and steam jet refrigeration system. Design and construction features as well as operating principles and characteristics are included and information is given on engineering considerations such as the calculation of capacity and the determination of air leakage in existing vacuum systems.

Steam Unit Heaters

No. 6882—Beacon-Morris Corp., 60 Brainerd Rd., Allston, Boston 34, Mass., has released a four page bulletin

G&O RADIATION

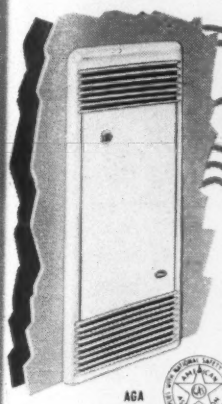


G&O COILS

FOR HEATING AND COOLING IN A WIDE RANGE OF SIZES

Send for Catalog

THE G & O MANUFACTURING COMPANY
New Haven, Connecticut
Pioneer Manufacturers of Square Finned Tubing in the United States



Money-Maker for Single Job or Big Contract

UTILITY

WALL-O-MATIC
built-in
Circulating Heater

AGA APPROVED

Here's a job-getting, contract-winning wall heater. It installs quickly, simply, at low labor cost. You can sell it as the sole heating equipment for apartments, smaller homes, offices, stores . . . and as an *extra* heat source for larger homes and areas. It's a "natural" for home subdivisions and apartment house developments.

The *Wall-O-Matic* is designed for all types of single and multiple-story structures, and all types of construction. Comes to you pre-assembled with plaster guides and header for easy installation in any standard 4" stud wall without furring. 15,000 to 50,000 BTU input, single and dual models, vented. Automatic or manual controls. Richly styled to lend beauty to any setting.

Write for Catalog Sheet and Complete Information

Line Up With the Utility Line of Heating, Cooling and Ventilating Appliances

Forced Air Furnaces
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Unit Heaters—Suspended Type
Evaporative Air Coolers
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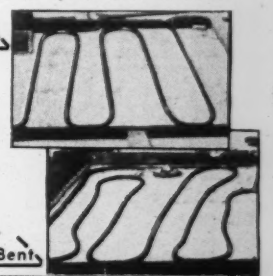
Announcing

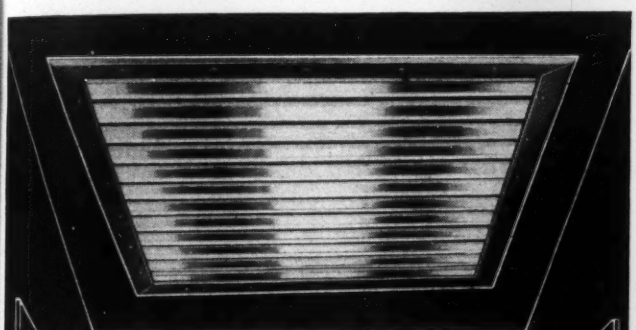
TAL Portable Tube Bender *for radiant heating*

Bend right on the spot, fast and accurately

Now, with this new light-weight machine that can be carried in one hand, you can speed-up radiant heating installations . . . for the Tal Tube Bender handles, without annealing, *hard copper* tubing as well as soft, both K and L, in all diameters from 3/8" to 1" in all standard radii. Changeovers from one size to another can be made in 40 seconds. One man does everything without help. No flattening, no kinks, no wrinkles. Made and guaranteed by Tal, the world's largest exclusive producer of pipe and tube bending equipment. Write today for data bulletin.



Tal Bender, Inc. Dept. 13 Milwaukee 2, Wis.



AIR-FLO

AUTOMATIC

CEILING SHUTTER
FOR ATTIC FAN

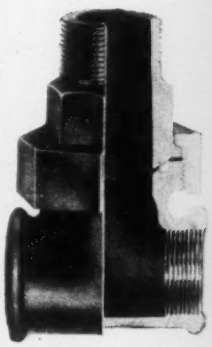
Built so they can be installed practically flush with the ceiling, AIR-FLO Ceiling Shutters present a refined, finished appearance. Their natural aluminum color blends with any decoration, eliminating need for painting, and no grille or winter cover is required. Furnished in 5 different widths, single panel up to 73" long. No operating mechanism shows. Built-in fusible link. Meets fire underwriters requirements.

WRITE FOR NEW CATALOG 43-D

Illustrations and details of the complete AIR-FLO line.

AIR CONDITIONING PRODUCTS CO.
2340 W. LAFAYETTE BLVD. • DETROIT 16, MICH.

Leakproof-Permanent
Low Cost-Fast Work



IT'S THIS EXCLUSIVE RECESSED BRASS SEAT that means JOINTS THAT STAY TIGHT!

. . . when you use

JEFFERSON SPECIALTY UNIONS

This exclusive feature provides virtually a one-piece permanent, leakproof construction because the recessed brass seat is integral with the iron of the fitting. This is accomplished by locating the seat away from the runway of the fitting and by grinding the seating surfaces together in pairs to assure a self-seating joint which can be made up *tight* with minimum pressure.

Jefferson Specialty Unions are made in all types for your piping needs. Please write us now for more detailed information.

JEFFERSON UNION CO.
607 West 26th Street, New York 1, N. Y.
35 Fletcher Avenue, Lexington 73, Mass.
65 Gooding Street, Lockport, N. Y.

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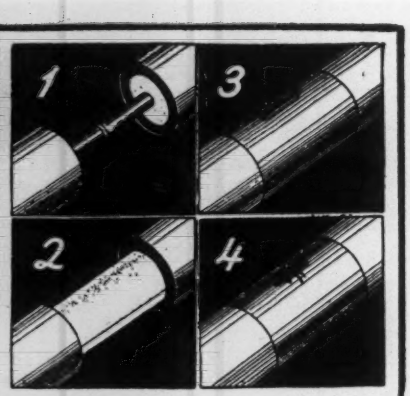


4 Simple Steps

1. Field joint ready for inspection. 2. Joint covered with standard pipe insulation. 3. Durant joint casing in place ready for asphalt. 4. Asphalt poured in slot making a perfect seal.

BUILT-IN at FACTORY

For all your underground and exposed piping jobs you'll like DURANT Pre-Sealed Insulated Pipe. It is waterproof . . . reduces installation costs to a minimum . . . eliminates electrolysis and corrosion . . . requires very little trenching and field work . . . avoids extra cost of waste and breakage . . . comes to you ready to install. In short, D.I.P. is the *perfect pipe* for this type of job. Investgate D.I.P.! Write for details today.

Ready to Install

PRE SEALED

FOR STEAM HOT WATER REFRIGERATION

Trade Mark

Reg. U.S Pat. Off

D.I.P.

Durant
INSULATED PIPE CO.
1015 Runnymede St.
Palo Alto, Calif.

The ORIGINAL Pre-sealed Insulated Pipe

on its series 200 steam unit heaters. Dimensions, capacities, prices, and other pertinent information are given on these units which incorporate vertical tubes of seamless copper brazed into seamless copper headers. Capacities range from 24,000 to 360,000 Btu per hr.

Steel Boilers

No. 6883—The line of type D steel boilers offered by Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York 17, N. Y., is described and illustrated in a 12 page catalog. Complete information is given on these boilers which are designed for use with oil, gas, or stoker firing and which are available in sizes from 2680 to 42,500 sq ft EDR steam.

Stokers and Oil Burners

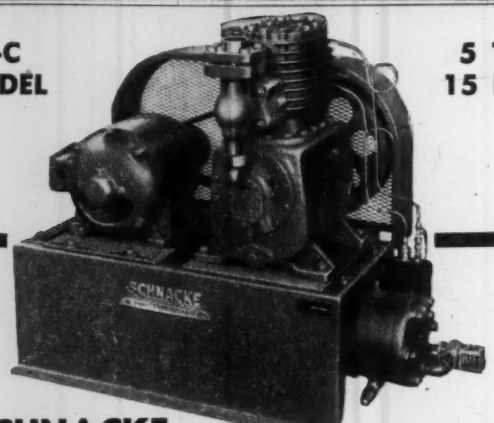
No. 6884—Commercial and industrial stokers and oil burners, together with the company's line of controls, are among the featured products described in a new large bulletin issued by Iron Fireman Co., 4784 S.E. 17th Ave., Portland, Ore. The bulletin, which covers seven lines of products, also includes descriptive material on residential heating equipment for coal, oil, and gas firing.

Tube Fittings for High Pressure Systems

No. 6885—The role of its tube fittings in the sealing of fluids at high pressures is covered in a new catalog issued by Flodar Corp., 331 Frankfort Ave., Cleveland, Ohio. These tube fittings, featuring an alloy steel spring sleeve, are described and illustrated, and design, construction, and performance features on high fluid pressure installations are covered.

T-C MODEL

5 TO 15 H.P.



SCHNACKE
COMPLETE
CONDENSING UNITS
For Air Conditioning and Refrigeration

SCHNACKE Compressor has replaceable, refrigerant-cooled steel cylinder sleeves, balanced forged crankshaft, sealed crankcase, complete pressure lubrication, and many other advanced features.

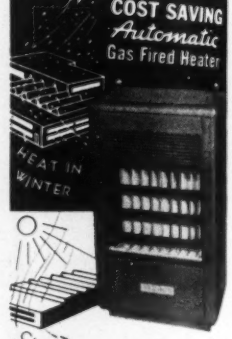
Condenser is ASME approved, fin tube, cleanable type, mounted in sturdy base. Water control valve and pressure control installed. F-C Models 10 to 25 H. P. available. Models less condenser, from 5 to 50 H.P.

Our new foundry now makes deliveries on most models possible in one to two weeks.

Write SCHNACKE, INC., 1020 E. Columbia St., Evansville, Ind., for complete details.

SCHNACKE, INC.

Specify
THE MODERN, TROUBLE-FREE UNIT HEATER



Year around utility, plus complete and *automatic* control of heating, feature Reznor gas fired unit heater installations. Specific areas can be comfort-regulated to requirements . . . no need to heat all of partially used buildings to bring heat to sections in use . . . just cut in Reznor units as needed.

EASY INSTALLATION

Installation is easy and fast. Install the heaters—pipe the gas—connect the electric service, and the best obtainable *automatic* heating system is ready for service.

Reznor heaters are designed to be used with natural, manufactured, and liquid petroleum gases. Write today

REZNOR SPECIAL HEAT EXCHANGER AND BIG, QUIET FAN DELIVER MORE HEAT FROM FUEL

SUSPENDED OR FLOOR MODEL
A SIZE FOR EVERY NEED

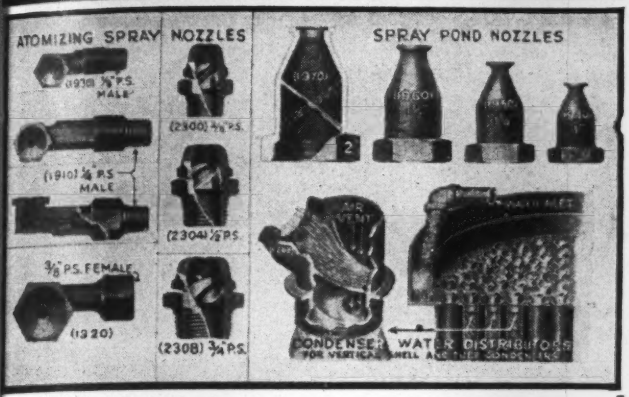
REZNOR
Gas fired unit heaters

REZNOR CO. MERCER 8, PENNA.
GAS FIRED HEATERS SINCE 1888

NO BOILERS • NO STEAM LINES
NO FUEL STORAGE • NO FIRE TENDING

QUALITY always wins!

MARTOCELLO Products are fully DEPENDABLE



ATOMIZING SPRAY NOZZLES, SPRAY POND NOZZLES, and CONDENSER WATER DISTRIBUTORS

For Vertical Shell and Tube Ammonia Condensers

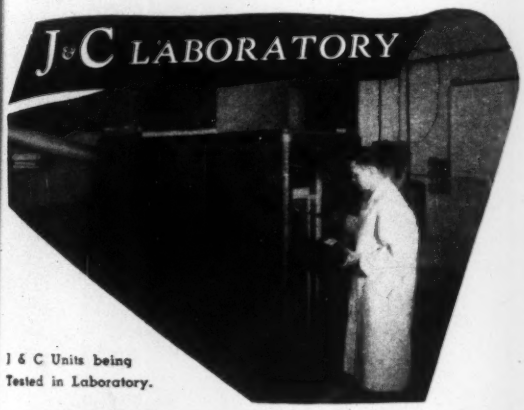
ALL MARTOCELLO SPRAY NOZZLES are manufactured with precision and of a design which has been thoroughly tested for maximum results and durability. They are guaranteed to give satisfaction. Successful, efficient results depend largely upon selecting the proper number, type and size of Nozzles suitable for your installation.

THE MARTOCELLO CONDENSER WATER DISTRIBUTORS are now used as standard equipment by Progressive Refrigerating Engineers in solving their Manpower problem, because they require no attention and assure users of the lowest Condenser Operating Pressures and Minimum Power Cost.

It will pay you to consult us. WRITE or WIRE for further information.

Jos. A. Martocello & Co. 229-231 NORTH 13th STREET PHILADELPHIA 7, PA

Serving the Ice Industry Since 1916" . . "There is no Substitute for MARTOCELLO QUALITY"



J & C Units being Tested in Laboratory.

See our CATALOG in SWEET'S FILE

PROVES PERFORMANCE

PRECISE MEASUREMENT . . . Of exact static pressure drop across each J & C unit and proper blower size for each J & C Model.

EXHAUSTIVE TESTS . . . of J & C units provide installation engineers with the data needed for correct installations.

ONLY IN THE LABORATORY . . . may technicians definitely establish:

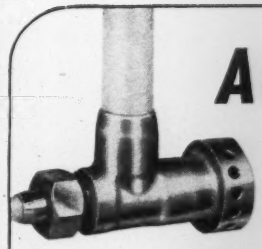
Correct Pressure Loss Measurements
Proper Temperature Rise
Accurate Flue Gas analysis
Known Heat Transfer

NO GUESSWORK . . . The J & C laboratory, working for you, permits accurate installations because performance capabilities are precisely determined.

THE COMPLETE LINE . . . . COMPLETELY PROVEN

J & C, America's largest and most complete Warm Air Heating Line, offers over 100 types and sizes with outputs from 3,800,000 down to 100,000 Btu in the Famous Tubular Series . . . other models down to 52,500 Btu. Exact engineering plus endless testing provide the J & C features that give you an "edge" when you specify or install J & C.

A PRODUCT OF
JACKSON & CHURCH COMPANY, SAGINAW, MICHIGAN



Automatic Liquid Feed without Moving Parts

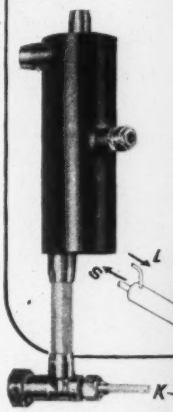
THE PHILLIPS INJECTOR provides completely automatic evaporator feed with *no moving parts*. By supplying recirculated liquid steadily to the evaporator it increases heat transfer and increases the evaporator capacity.

Many existing dry-expansion evaporators can be easily changed to Injector operation, with improved performance assured.

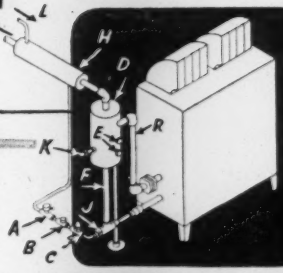
Adaptable to "Freon" applications, from air conditioning and water chilling to cold storage installations with plate coils, fin coils, and unit coolers. Injector capacities from fractional to 200 tons with "Freon". Furnished with either single outlet or distributor head.

Write for sketches showing typical Injector installations.

Injector with Distributor Head



Injector and Surge Drum Assembly



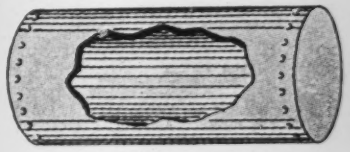
FLOOR TYPE ROOM COOLER HOOK-UP

A—MAGNETIC SOLENOID
B—BULLS-EYE
C—HAND VALVE
D—SURGE DRUM
E—SIGHT GLASSES
F—LIQUID LEG
H—HEAT EXCHANGER
J—INJECTOR
K—CHARGING CONNECTION
L—LIQUID LINE
R—GAS RETURN
S—SUCTION LINE

H. A. PHILLIPS & CO.

3255 WEST CARROLL AVENUE • CHICAGO 24, ILLINOIS

NEW LIFE!



to Old Water Tanks

WITH APEX PERMANENT PORCELAIN LINING. Pitted or leaky water tanks can be restored to original usefulness by relining, applied in two coats approximately ¼" thick—cures under water—has an affinity for lime—prevents corrosive and pitting action on metal—withstands temperatures up to 250° without cracking—adds years to life of tank.

RIDAL SERVICE CLEANS

CHEMICALLY STEAM & WATER SYSTEMS. Eliminate sludge and foreign matter from your entire plant safely and economically. We clean and restore to original operating efficiency CHEMICALLY such equipment as: TURBINES — CONDENSERS — STEAM FEED LINES — CONDENSATE RETURNS —WATER HEATERS—STEAM KETTLES—COOLING AND HEATING COILS — PREHEATERS — HEAT EXCHANGERS — EVAPORATORS — TRAPS OF ALL KINDS — ETC.

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